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The question of derailments,

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The stages of development of the railway, like in general all technical applications based on new principles, can be taken as following each other in the following order :

a) first of all the railway, including both track and rolling stock, was built by trained men of good practical sense; then when these had been developed, it was seen that mistakes had been made here and there, so that

b) the causes of these mistakes were looked into. This implied finding out the theoretical foundations of the mistakes and, there being no theory, one had to be created.

Moreover, as regards the technical problems of the railway, it is necessary to consider the influence of numerous factors each of which must be judged separately from the theoretical point of view. Since owing to the great number of factors and of divers circumstances, theory has only followed practice very slowly; the result is that up to the present in railway technique, with a few exceptions, the laws are all inspired by practice. It is natural not to wait for things to get worse. When one is close to achieving the best practical results compatible with circumstances, it is impossible to progress any further without the basis of an acceptable theory. We think that this is the point at which the railways have marked time during the last few years. Owing to the need to increase progressively the speeds and obtain the most economical working, the most pressing question is to perfect the theories based on practical results. This work will still take a great many years. We are taking the liberty of adding our small quota, making it quite clear that its only object is to signpost an intermediate stage from which perhaps the final objective may be sighted.

The question of the relations between the rail and the vehicle, and in consequence, that of derailments, is one of the subjects under study in recent years. In addition, the railways and railway associations have been carrying out trials on this subject.

Derailments can be due to a great many factors. The following are mentioned as examples :

— faulty construction of the track or rolling stock, for example wheels of too small a diameter for the track in question;

— faulty maintenance of the track and rolling stock, for example subsidence of the track or a broken axle;

— operating mistakes, for example a mistake in working points and crossings, an obstacle on the track, or the shifting of the load on a wagon.

If a single one of these causes is selected, for example a shifted load, to carry out a complete analysis of the phenomenon of

derailment which results therefrom, the result is a checkmate. There is nothing else to be done but to simplify the problem. This is the only way in which the most essential factors can be examined and then linked together into a system.

CHAPTER I.

We will leave aside such obvious faults as a broken axle. We will look into the essential reasons which enable vehicles to roll and slide with safety over rails. This safety depends on the flange (which is less than 40 mm high), the dimensions of which were established after long years of practical experience. For this reason, there is no cause for astonishment that these dimensions are specified exactly in millimetres. The safety of the passengers comfortably travelling in the express coaches as well as that of the goods sent by rail both depend on this little flange. If the outside of the flange jumps up to the same level as the top of the rail, a derailment may occur. First of all, to simplify our calculations, we will leave aside the lateral stresses and take it that the height of the flange is based solely on safety considerations. Further on, when we will be dealing with many factors, we will abandon this simplified hypothesis which does not correspond to the real position.

Let us take as a basis the most widely used type of vehicle in railway operating and the most suitable for our study, the four wheeled vehicle. We will take this as having outside springs and a completely rigid frame. By this rigidity we mean that it is impossible to lift one corner of the frame around the diagonal of the frame Such in general are the vehicles which have a short wheelbase and tank wagons in which the tank and its carrying members form a very rigid whole.

Apart from this, we will take the weight as being symmetrically distributed.

The points of contact of three wheels of the vehicle form a plane. We will now try to drive down below this plane the part of the rail lying under the fourth wheel. If this experiment is carried out, it will be found that the vehicle after a certain limit will only rest on three of its wheels. Let us ascertain where this limit lies.

The state of equilibrium of the forces bearing on the axle in this position of the vehicle is shown in figure 1. We will leave out the small degree of asymmetry due to the displacement of the centre of gravity.

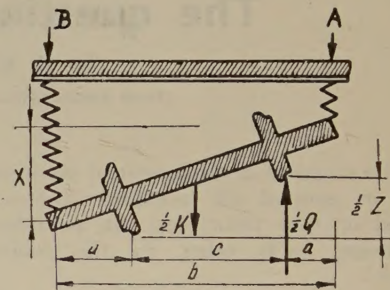


Fig. 1.

Notations :

tare of the vehicle	Q kg
weight of the non-suspended parts.	K kg
suspended weight (Q-K)	P kg
specific deflection of a spring . .	f mm/kg (right-handed characteristic)
reactions of the springs	A and B kg
distance of the circular tracks . .	c mm
distance between centres of the springs	b mm ($2a = b - c$)
difference in deflection of the two springs	X mm
difference in level of the two wheels	$1/2\ Z_1$ mm

There will be equilibrium if

$$A = \frac{1}{2} Q \frac{a + c}{b} - \frac{1}{2} K \frac{a + \frac{c}{2}}{b}$$

and if

$$A + B + \frac{1}{2} K = \frac{1}{2} Q$$

The difference between the deflections o the two springs is

$$X = (A - B) f$$

The wheel on the side of the force B begins to lift when it is no longer transmitting any pressure on the rail. This is the situation shown in figure 1.

The drop in one wheel compared with the other is :

$$1/2\ Z_1 = X\frac{c}{b}$$

From the above, we conclude that the wheel will leave the rail if the latter falls away below it by at least

$$Z_1 = f\left(\frac{c}{b}\right)^2 Q \qquad \text{(Formula 1).}$$

At this stage, around the diagonal joining the two springs which receive most of the force, the frame itself will rock, since one corner is no longer supported. However, the spring opposite the other diagonal is still loaded. The body will therefore rock until this opposite spring is unloaded. There must therefore be a drop in the rail of $2\ Z_1/2$ for the wheel to leave the rail. In this case, beyond this position, the vehicle can already rock. It is remarkable that in this unstable position, the vehicle only rests on three wheels, but the frictions which occur on the wheel flange may result in it only being supported by two wheels (naturally those diagonally opposite each other) while the other two diagonally opposite wheels are up in the air. This phenomenon can easily be seen on a badly built vehicle, with a gauge of 760 mm on running off a super-elevation. The case of the rocking vehicle can always be reproduced at this point, and the vehicle frequently is derailed when it is empty.

The lower the value of Z_1 on a vehicle, the greater the danger of derailment; the flange can jump over the rail more easily, i.e. over the top of the low point in the rail. Let us see what can be concluded from this simple formula.

From the derailment point of view, a vehicle is the more dangerous as :

a) its tare is lower. Let us recall certain regulations of the International Rules for Coaches (R.I.C.) which lay down the minimum tare for coaches to be used on express trains. We can infer from this that an empty vehicle is more easily derailed than a loaded one.

After they have been built vehicles must be taken on a trial run in order to make sure that they run safely. It should be noted that the trial run is made empty, because if there is any fault in the construction, it will be more easily appreciated when the vehicle is empty. (It is for another reason that it is forbidden to load vehicles with bearings until the bearings are polished and our remark about trials unloaded will also refer to vehicles fitted with roller bearing boxes.)

b) as the value of c/b is the lower. This is one of the causes why narrow gauge stock is more easily derailed than standard gauge stock. On narrow gauge lines the tare is lower, and the square root of the ratio c/b is around 0.35 compared with the value 0.56 for standard gauge stock; the flexibility « f » of the springs is also lower, at the most equivalent to that of the standard gauge; finally as regards the construction and maintenance of the track there are more low places in narrow gauge lines than on standard gauge lines.

c) as the value of f is lower, i.e. the harder the spring of the vehicle. This is the reason why wagons are derailed more easily than coaches. (It would be an advantage in the interests of the safety of the passengers to build coaches which will not be derailed should a spring break. We will return to this point further on.)

At the beginning, we considered a vehicle with the load removed from one wheel only owing to a lowering of the rail. Does such a case occur in practice? This phenomenon

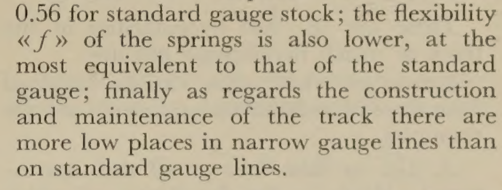


Fig. 2.

always occurs on superelevation gradients and if the rails are lower on one side. Let us take a vehicle seen from the side placed on a superelevation gradient.

The superelevation gradient is usually fixed at least at 1/400. If the distance between centres of the axles is t, we have lowered the rail under a wheel by $t/400$. If this value is equal to Z_1, or if it is greater, there is danger of derailment.

The greatest drop in the rail, even on the worst lines, cannot exceed 30 mm (1 3/16″).

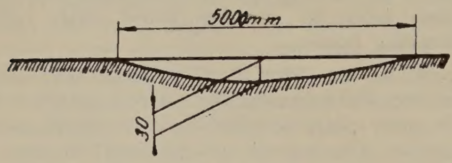


Fig. 3.

These are the two irregularities in the track to which derailments are due.

Let us see what values can be obtained from these considerations applied to concrete examples.

A. Let us take first of all this narrow gauge vehicle in the derailment of which we are interested.

The characteristics of this vehicle are :

tare	Q	5 000 kg
non-suspended weight . .	K	900 kg
suspended weight	P	4 100 kg
flexibility	f	0.016 mm/kg
distance between centres of the axles	t	4 000 mm
	c	806 mm
	b	1 410 mm

With these values, $Z_1 = 26$ mm. At the point of derailment, the down gradient of the superelevation was equal to 1/300. The depression was not measured separately. Below the wheels of the vehicle, there was therefore only a difference of 4 000/300, i.e. 13.3 mm. Unevennesses in the construction of the vehicle, rubbing between the blades of the suspension springs and other frictions in the suspension gear greatly diminish this theoretical value of Z_1, below which the wheel will not come down, even if the rail sinks still more.

B. As our second example let us take a light tank-wagon with four wheels, for standard gauge lines. Let us suppose that it has the following characteristics :

tare	Q	8 000 kg
non-suspended weight . .	K	2 500 kg
suspended weight	P	5 500 kg
	f	0.01 mm/kg
	t	5 000 mm
	c	1 500 mm
	b	2 000 mm

From these figures we get :

Z_1 . . . . 45 mm

The superelevation gradient is equal to 1/400. The difference in height under the wheels is 5 000 : 400 = 12.5 mm. This shows how much safer a standard gauge vehicle is.

I. a.

Let us now examine by these simple means the case occurring when one of the four springs breaks. Figure 4 shows what happens. Owing to the complete symmetry, the reactions of the springs diagonally opposite each other are always identical. Consequently

$$A = C \text{ and } B = D.$$

The full load is equal to Q :

$$Q = 2 \ (A + B).$$

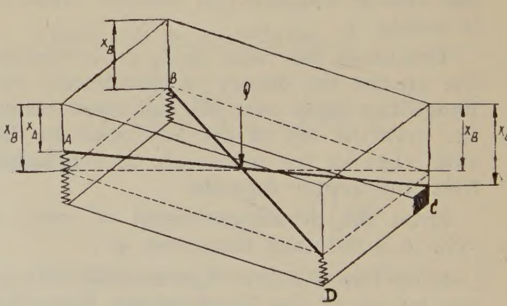


Fig. 4.

The figure shows the body dropped. If the four springs were equally loaded, each of them would only have deviated from their non-loaded position by X_B, i.e. they would take up a parallel level a little higher up. Owing to the broken spring, the drop at B from the starting position would be X_B.

$$B = D = \frac{X_B}{f};\ A = C = \frac{X_A}{f};$$

$$Q = 2\left(\frac{X_A}{f} + \frac{X_B}{f}\right);\ X_C = X_B + (X_B - X_A),$$

whence

$$X_B = \frac{1}{3}\left(\frac{Q}{2}f + X_C\right) \text{ and } X_A = \frac{1}{3}(Qf - X_C)$$

$$X_B - X_A = \frac{2}{3}\left(X_C - \frac{Q}{4}f\right);$$

$$Z_2 = \frac{c}{b}\frac{2}{3}\left(X_C - \frac{Q}{4}f\right) \qquad \text{(Formula 2).}$$

Naturally it follows from what was said on page 3, that the drop in the wheel would be less than $(X_C - X_A)$; it will be reduced in the proportion of c to b.

Here we also see that the lower the value of Q, the tare plus the load of the vehicles, the greater the unloading of each spring. The greater is the value of X_C, i.e. the greater will be the proportion of suspended weight, and the greater will be the unloading. The safety brackets serve to limit the displacement of the suspended part. If a spring breaks, the two parts come into contact, the unloading is less and the danger of derailment is lessened.

These safety brackets should be placed above the spring buckle in such a way that there can be no contact due to the normal oscillations in working; the construction and assembly tolerances must also be taken into account, as well as other factors such as dissymmetric loads. If the position of the brackets has been designed with these considerations in mind, if greater oscillations occur the spring buckle will be struck and the result will be that the unloading of the springs will be reduced to the smallest possible value.

I. b.

What happens if a spring is above the plane determined by the other three? This is the same case as that of a spring having a greater camber than the others.

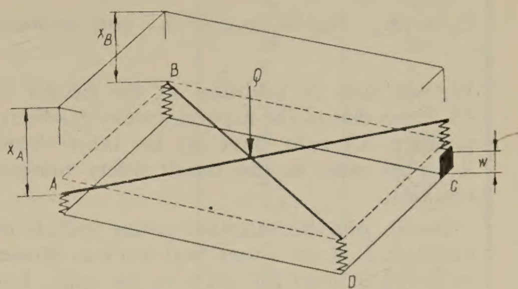


Fig. 5.

Let us suppose that the error in assembly is of a value w. According to the symmetry

$$A = C = \frac{X_A}{f};\ B = D = \frac{X_B}{f};$$

$$Q = 2\left(\frac{X_A}{f} + \frac{X_B}{f}\right).$$

If the load is allowed to weigh progressively on the mechanism shown, first of all it will drop $1/2\,w$ mm whilst only loading springs A and C. The springs B and D will only come into play subsequently. After this, all the springs will take up an equal share of the load. The corresponding drop has the value

$$X_A - \frac{w}{2}.$$

So that :

$$X_A - X_B = \frac{w}{2};$$

$$Z_3 = \frac{c}{b}\,\frac{w}{2} \qquad \text{(Formula 3).}$$

Naturally the value has to be brought back from $X_A' - X_B$ to the gauge of the wheels, i.e. be multiplied by c/b.

Let us suppose that a spring of a narrow gauge vehicle dealt with in example I. *a*, has a camber of more than 10 mm above that of the other springs and at the same time is fitted 10 mm above them. Then $w = 20$ mm and

$$Z_3 = \frac{c}{b}(X_A - X_B) = \frac{c}{b}\, w\; 1/2 = 5.7 \text{ mm approx.}$$

We see that in addition to the height of 13.3 mm due to the superelevation gradient, another 5.7 mm have to be taken from $Z_1 = 26$ mm, so not much safety margin remains.

Let us put the standard gauge vehicle of example I. *b*, on a very bad track as shown in figure 3, over the drop in the rail. Let us take w as equal to 20 mm as above.

$$Z_3 = \frac{c}{b}\,\frac{w}{2} = 7.5 \text{ mm.}$$

In this case, the drop takes 30 mm from Z_1 which equals 45 mm. After taking away the 7.5 mm as calculated above, there is not a great margin of safety in this case either.

Owing to the inclination of the frame measured by the values $X_A - X_B$ determined in paragraphs I. *a* and I. *b*, of the original value of Z_1 only the height of $Z_1 - Z_2$ remains, i.e. $Z_1 - Z_3$ to prevent derailment owing to irregularities in the track, etc.

I. c.

However, there is another factor which comes to the aid of the springs to prevent derailment. This is the torsion of the frame, or if there is a body, the torsion of the two together.

To measure this torsion of the vehicle, one corner of the vehicle is raised by one millimetre and the force needed to do this is measured. During this time, the three other corners of the vehicle remain in their original plane. Here, by « corner of the vehicle » is not to be understood the extreme angle of the frame, but the point of attack of the suspension spring. The following table based on trials carried out, refers to this factor of torsion. It can be adapted to other vehicles according to their dimensions (see *Bibliography*, No. 2 and Table I hereafter).

This table shows that the increase in the distance between centres of the axles make the vehicle easily affected by torsion; the body makes it rigid and the partitions in the body, as in coaches for example, make the vehicles still more rigid. Welded construction also increases the rigidity.

Let us designate by ψ the inverse value of the factor of torsion. Its unit of measurement is the same as that of f. Let us repeat figure 1 but now considering it from the point of view of the torsion (fig. 6). In the figure, the forces shown below and above the springs do not appear to be in equilibrium. However, the movement « M » showing the other half of the vehicle (divided along the diagonal) gives perfect equilibrium.

The deflections of the springs are

$$Af = a' \text{ and } Bf = b'$$

The drop due to torsion is equal to

$$\left(\frac{Q-K}{2} - B\right)\frac{\psi}{2} = X.$$

Here, only half the value of ψ must be taken, because in getting out the figures given in Table I, only one corner of the frame was raised, whereas the two opposite corners are concerned.

Let us designate by L the height of the springs when free. Then

$$= X_1 + (L - b') - (L - a');$$

$$A + B + \frac{1}{2}K = \frac{1}{2}Q$$

$$Ab = \frac{1}{2}Q\,(a + c) - \frac{1}{2}K\left(a + \frac{c}{2}\right);$$

TABLE I

Series mark	DESCRIPTION	Axle wheelbase	Torsion factor in kg/mm
	a) Two wheeled vehicles.		
Om	Rivetted, built in union	4.5	7-14
Omm	Welded, with reinforced sills, and outside members	6	135-250
Ommu	Welded, without reinforced sills	5 3/6	12-22
Sm	Rivetted	8	3-5
Sm	Welded	8	5-7
Sm	Welded, with outer sills	8	30-38
R	Rivetted	7	4-9
R	Welded	7	8-10
Rs	Welded, with reinforced sills	8	33-55
Gr	Rivetted	4.5	53-90
Ghs	Rivetted (Kassel)	5.3	110-163
Ghs	Rivetted (Kassel)	6	70-110
Ghs	Welded (Oppeln)	7	32-90
Glhs	Welded (Dresden)	7	53-220
Gkhs	Refrigerator vehicle welded in one with the body	7	360-600
V	Rivetted	4.5	125-280
V	Welded	4.5	360-420
Tank-wagon	Rivetted	3.9	510-540
»	Rivetted	4.5	320-480
»	Welded	4.5	820-1 000
Pwgs	Welded van for high speed goods trains . . .	6	570-660
»	Like the former, but with metal body	7	360-400
»	Like the former, but with wooden body . . .	7	60-80
	b) Four wheeled coaches.		
Bi	Built in 1928 with rivetted metal body	8.5	360-495
Bci	» » »	8.5	260-300
Ci	» » »	8.5	495-550

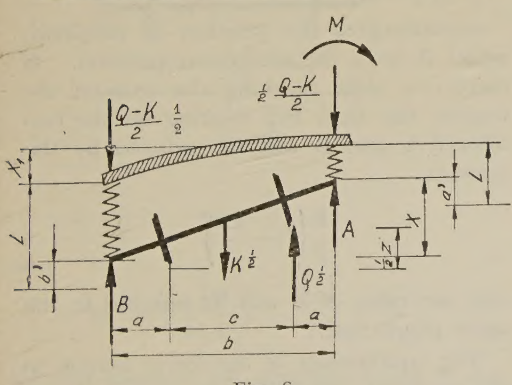


Fig. 6.

$$\frac{1}{2} Z_4 = \frac{c}{b} X; \; a = \frac{b-c}{2}; \; a' = Af;$$

$$b' = Bf; \; X_1 = \frac{1}{2}\psi\left[\frac{1}{2}\frac{Q-K}{2} - B\right]$$

and from these formulae, we get :

$$Z_4 = \left(\frac{c}{b}\right)^2 Q\left(f + \frac{\psi}{4}\right) \quad \text{(Formula 4).}$$

The only difference compared with Formula 1 is that instead of f (lowering of the spring) we have to put

$$f + \frac{\psi}{4};$$

so that the torsion lessens the tendency to derailment.

In the numerical example given in paragraph I. *b*, we must add to a value of f of 0.01 mm/kg one quarter of the reciprocal value of approx. 500 kg/mm. This only represents 5 % of f, which is negligible. Consequently, in the case of tank-wagons and coaches, we can count upon a rigid frame, whereas with flat and hopper wagons we cannot neglect the torsion.

As an example, let us make these calculations for a hopper wagon, or else a covered wagon — the characteristics of the two are practically identical — both of which are exposed to derailment. Let us take the welded series Om and Gr vehicles from Table I. One has a tare of about 7.5 t and the other of 10 t. The flexibility of the springs is $f = 10$ mm/t.

In the case of the hopper wagon

$$Z_4 = \left(\frac{c}{b}\right)^2 7.5 \left(10 + \frac{1}{4 \times 0.01}\right) = 262.5 \left(\frac{c}{b}\right)^2;$$

in the case of the covered wagon

$$Z_4 = \left(\frac{c}{b}\right)^2 10 \left(10 + \frac{1}{4 \times 0.07}\right) = 135.7 \left(\frac{c}{b}\right)^2.$$

Consequently, from the point of view of derailment, the covered wagon is the more dangerous, whereas the Omm vehicle with reinforced sills is more dangerous than any of the covered wagons, because it has a more rigid construction and its tare is lower. We can also see that the frame of a tank-wagon is so rigid that its deflection is negligible compared with that of the springs. So there is all the more reason to leave out the torsion of the frames of coaches whose springs are still more flexible.

I. d.

Let us now consider what modifications the value of Z will undergo if the load shifts on a wagon or has been badly placed to start with. Here, it is necessary to take into account the fact that the possibility of resting on the two wheels stops completely and its place is taken by support on three wheels. From the derailment point of view, one of the four wheels will always be the most dangerous, and naturally we will only consider this latter (see fig. 7).

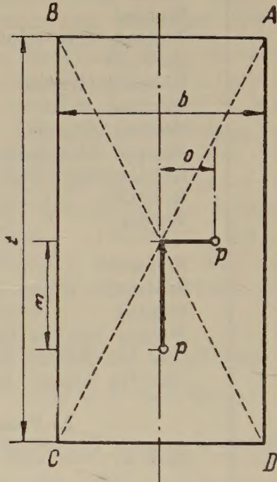


Fig. 7.

According to the position of the load, wheel B is in the dangerous position. If there is a shift m along the axis of the wagon, the load P/2 bearing on the two wheels A and B will be reduced by the amount

$$\frac{P}{2}\left(1 - \frac{2m}{t}\right)$$

and the value of Z will be reduced in the same proportion.

The equilibrium of the forces acting on the mounted axle A-B is shown in figure 8.

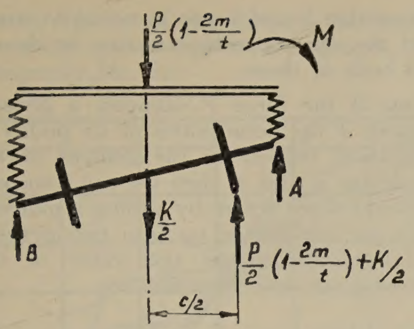


Fig. 8.

There is still here a free moment, the value of which is

$$M = \frac{c}{2}\left[\frac{P}{2}\left(1 - \frac{2m}{t}\right) + \frac{K}{2}\right].$$

Apart from this, the springs of the other mounted axle (C-D) are in equilibrium; but naturally this only occurs when the frame undergoes torsion. The amounts to be taken into account according to figure 9 are :

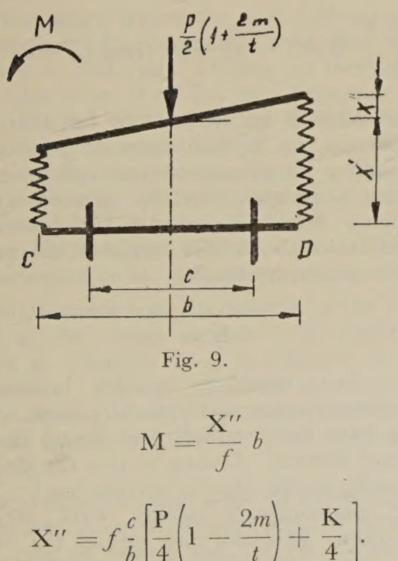


Fig. 9.

$$M = \frac{X''}{f}\, b$$

and

$$X'' = f\,\frac{c}{b}\left[\frac{P}{4}\left(1 - \frac{2m}{t}\right) + \frac{K}{4}\right].$$

So that between the supports A and B the torsion of the frame is : 2 X''. In addition, half of the value of Z_1 must also be taken into account.

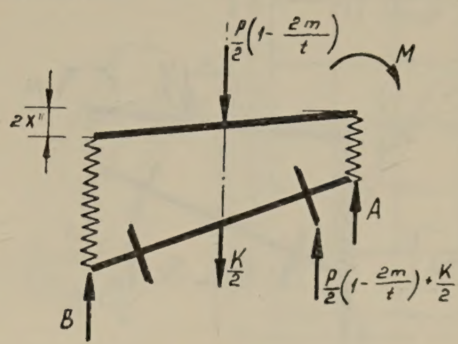


Fig. 10.

Consequently, as will be seen in figure 10

$$Z_5 = 2\,X''\frac{c}{b} + \frac{1}{2}Z_1$$

and here, in the formula of $Z_1/2$ instead of Q/2, we must put

$$\left[\frac{P}{2}\left(1 - \frac{2m}{t}\right) + \frac{K}{2}\right].$$

which brings us to the result

$$Z_5 = f\left(\frac{c}{b}\right)^2\left(Q - 2\,P\,\frac{m}{t}\right) \text{ (Formula 5).}$$

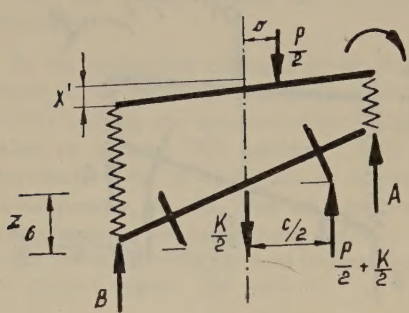
This shows us that compared with the original value of Z_1, the quantity Z_5 diminishes progressively as m increases.

We can also arrive at this formula by replacing the force P in figure 7 by an equal force applied at the centre of the wagon and a couple of forces of moment Pm. The force applied to the centre produces an increase in the value of Z_1, whereas the couple reduces it by

$$\left[f\left(\frac{c}{b}\right)^2 \; 2\,P\,\frac{m}{t}\right].$$

Now, if the force P of figure 7 is displaced transversely by the amount σ, then the

load on the two axles will remain the same, but the value of the non-equalised moment of the axle A — B has the value shown in figure 11



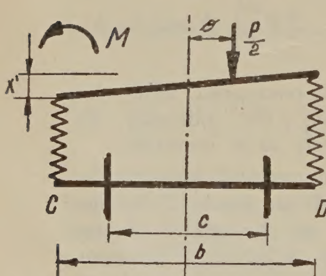
$$M = \frac{P}{2}\left(\frac{c}{2} - \sigma\right) + \frac{K}{2}\,\frac{c}{2}$$

in which

$$M = \frac{c}{2}\,\frac{Q}{2} - \frac{P}{2}\,\sigma.$$

Fig. 11.

On the other axle, there is a different deflection of the two springs, due on the one hand to the moment M, and on the other to the excentricity of the component P/2. The result is the torsion of the frame shown in figure 12.



$$M_ö = M - \frac{P}{2}\,\sigma$$

$$M_ö = \frac{c}{2}\,\frac{Q}{2} - P\,\sigma$$

$$\frac{X'}{2f}\,b = M_ö$$

Fig. 12.

On the wheel adjoining point C the levelling out until the load is right off will be

$$Z_6 = f\left(\frac{c}{b}\right)^2 \left(Q - 2\,P\,\frac{\sigma}{c}\right) \qquad \text{(Formula 6).}$$

Formulae 5 and 6 are identical in structure; the same conclusions can be drawn from both of them.

Now if the force P occupies a general position, if the coordinates of its points of application related to the centres of the vehicle are m and σ, then we can transpose this force to the centre by adding a moment which we can divide up into two perpendicular moments and the value of the flattening out will therefore be

$$Z_7 = f\left(\frac{c}{b}\right)^2 \left(Q - 2\,P\left[\frac{m}{t} + \frac{\sigma}{c}\right]\right) \quad \text{(Formula 7).}$$

For example let us examine what fault in construction will give rise to the same amount of flattening out of Z_1 being lost as in example A of the First Chapter owing to the effects of the superelevation gradient.

Of the value of Z_7, we are only concerned with the second term and with the data given in example A we get :

$$13.3 \text{ mm} = f\left(\frac{c}{b}\right)^2 .\ 2\,P\left(\frac{m}{t} + \frac{\sigma}{c}\right)$$

$$= f\left(\frac{c}{b}\right)^2 .\ 2\,PU = 42.9\,U\text{; then } U = 0.31$$

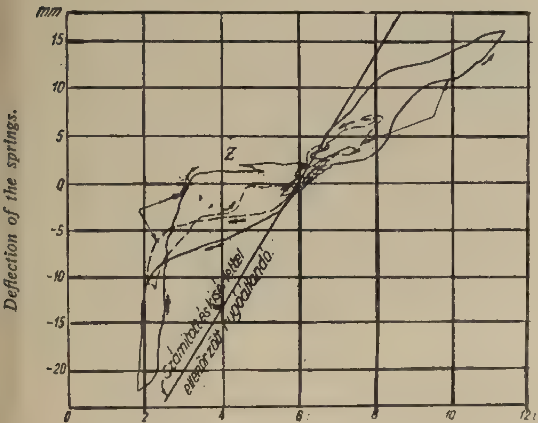
If the body is badly built, if for example $\sigma = 25$ cm, i.e. if the centre of gravity of the body is 25 cm out of line transversally, we lose as much as by the superelevation gradient. In the longitudinal direction, it would have to be 124 cm out of centre to give the same result.

I. e.

On four wheeled vehicles laminated suspension springs are generally used. The latter have the peculiarity of giving rise to internal friction. Owing to this the deflection only begins after a certain load. The same phenomenon occurs with weight reduction. This also may reduce the value of Z_1 by a few millimetres.

Figure 13 shows the effect of the simul-

taneous friction of the springs, the fastenings of the springs and the axle guard. (See *Bibliography* No. 3.)



Pressure of the springs plus half the weight of the pressure of the wheel.

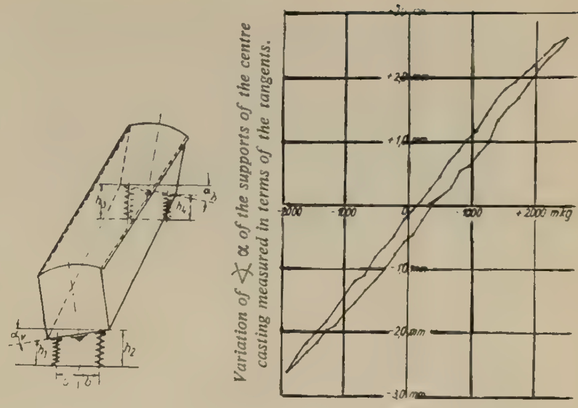
Fig. 13.

Translation of the Hungarian terms of the figure : Constant of the springs calculated and recorded on the trial engine.

This figure shows the real deflections of the springs of a locomotive with their loads during a trial run. During an increase of the load from 3 to 7 t the deflection is a bare 3 mm. Naturally the fact that on a locomotive several springs are connected together by compensation gear comes into this, and the result is that they are less sensitive. It is true that this does not occur on transport vehicles. The effect on locomotives is all the more striking.

In the same way, the torsion of the frame and of the whole vehicle with which we dealt in Chapter I. *c*, is affected by this friction. Figure 14 dealing with this phenomenon shows the results of measurements taken on a frame. (See *Bibliography* No. 4.)

We might also deal with the extraordinary cases which occur in practice, for example the case in which the specific deflections are not equal, the usual case on vans, the case in which the frame has been deformed



Variation of the moment and torsion in kgm.

Fig. 14.

after construction, etc. But this will not take us any further towards the solution of the essential problem. We will merely be straying from our objective.

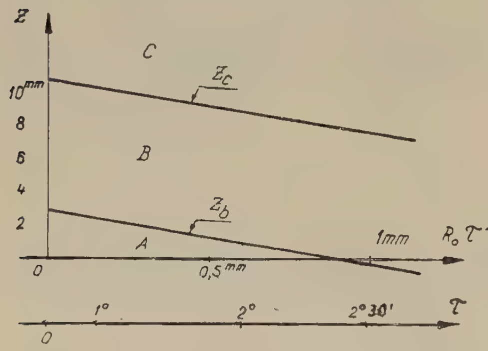
To sum up, it may be noted that all the vehicles have as characteristic of the deflections of the springs a value Z_1 or Z_4. The greater these are, the less the danger of derailment. Their value may be reduced by the quantities Z_2, Z_3, or the second member of Z_7, etc., and the result is to increase the danger of derailment.

CHAPTER II.

Up to the present we have only considered the vehicle while at rest. But the vehicle in movement will be more dangerous from the point of view of derailment if the wheel under which the rail drops is at the same time the driving wheel. In this case, there are also transversal forces which act upon the flange and mean that it is no longer necessary for the load to fall to zero under the wheel before derailment takes place.

The research work carried out to date on this subject is briefly summed up by Mr. CHARTET who, adding thereto the results of his own studies, has arrived at a result which brings out the phenomenon more clearly. In his article will be found

all the data needed, so we will not go into it in detail. (See *Bibliography* No. 5.) On the basis of these studies we can add to the values of Z calculated in the preceding chapters, the height of the conical part of the flange, i.e. a certain fraction of the height, known as « *ny* », depending on the angle of attack, because at this point the value of the transversal force needed for derailment is constant and at the maximum.



for R_0 = 500 mm.

Fig. 14*a*.

Figure 14*a* reproduced above is figure 6 from Mr. CHARTET's article.

According to the investigations of Mr. CHARTET, the displacement of the load from the wheel leaving the rail onto the other wheel increases the danger of derailment to an extent which can be expressed by the following formula :

$$\frac{Y}{R} = k_1 - \frac{k_2}{R}\,\frac{Q}{4} \qquad \text{(Formula 8).}$$

R denotes the real load of the wheel leaving the rail;

Q/4 denotes the nominal load of a wheel, i.e. one quarter of the total weight of the vehicle;

k_1 and k_2 are constants which, according to Mr. CHARTET can be taken for a conicity of the tyre of 70° as equal to $k_1 = 2$ and $k_2 = 0.7$ if $\varphi = 0.2$ and $\varphi' = 0.3$ (probable value on a dry rail).

The values of Y/R are shown in the diagram, figure 15.

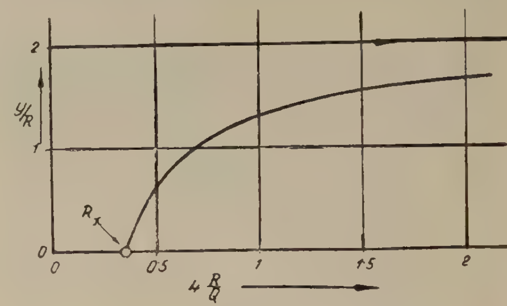


Fig. 15.

It will be seen that it is not possible to load the « derailing » wheel up to more than a limiting value

$$4\,\frac{R_x}{Q} \quad \text{(in hundredths)}$$

otherwise the wheel from which the load has been removed would leave the rail without any transversal force. In the example illustrated in the figure, the limit would be

$$R_x = 0.35\,\frac{Q}{4}.$$

It must be noted that this value has not been determined from trials but has been obtained by extrapolation. It can however be used as a basis for making studies.

For the sake of completeness, it should be pointed out that here we come to the limit of the zones B and C of figure 14*a*, i.e. the rounded part of the flange has not yet jumped over the head of the rail. For it to do so, the wheel must lift still more and so the spring must be deflected still more which presupposes an increase in the load on the wheel. However, in zone C, the value of the transversal force which can be allowed on the mounted axle begins to fall suddenly. There is no need to take this zone into account as in practice it does not give any appreciable value.

Finally, we can go on to the stresses affecting the axle in the transversal direction, i.e. horizontal stresses, which also may lead to derailment. In general, all the horizontal stresses which tend to derail the guiding wheel increase at the same time as the load on this wheel. For this reason, it can be understood that a transversal stress protects the more against derailment as it raises the load on the wheel. This can easily occur, for example close to the value R_x.

Let us designate as H the increase in the load on the wheel due to the transversal stress Y. From what has been said above we know that the load on the wheel must not fall below

$$R_x = \frac{k_2}{k_1} \cdot \frac{Q}{4} \qquad \text{(Formula 9)}$$

even if there is no transversal stress. Let us now examine the limiting value above which the horizontal stress (characterised by Y/H) gives warning of the danger of derailment. We can note that

$$\frac{Y}{R_x + H} = k_1 - \frac{k_2}{R_x + H} \cdot \frac{Q}{4}$$

and if we take formula 9 above into account giving R_x, we get as a result

$$\frac{Y}{H} = k_1 \qquad \text{(Formula 10).}$$

We can therefore avoid derailment if we keep the load on the wheel below R_x and if in addition the value of the transversal stress Y/H remains below k_1.

What happens when there are several horizontal stresses acting on the axle? Let us consider the case in which the wheel is already carrying a load « R » which can fall from Q/2 to R_x, and let us suppose that this wheel is in addition submitted to horizontal stresses Y_i which cause additional increases H_i in the load on the wheels. Let us suppose that each of the stresses is such that in itself it will not cause derailment, i.e.

$$Y_i/H_i \leqslant k_1.$$

If this were not so, this stress by itself would derail the vehicle, so that there would be no point in continuing our examination. Our formula 8 can be written in a simpler form :

$$Y = k_1 R - k_2 \frac{Q}{4} \qquad \text{(Formula 11).}$$

According to this formula, we can represent Y in terms of R by figure 16.

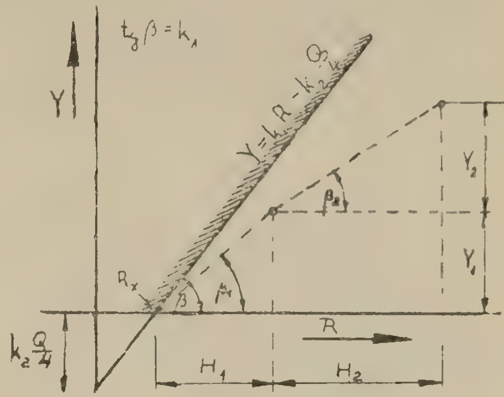


Fig. 16.

It will be seen that it is sufficient to try not to exceed the line representing Y, because the axle cannot support a greater transversal force without derailment. If we do not go below R_x for the load on the wheel, we should not exceed the value of $Y_1/H_1 = \tan\ \beta = k_1$, in order to keep above this line. Now, if we have several similar stresses, for example, as the figure shows, Y_1/H_1 and Y_2/H_2, and if the prescriptions of formula 10 are observed in each case, we will never come within the danger zone; on the contrary, our margin of safety, i.e. the value of the force Y still admissible relative to the point R_x will continue to increase. Returning to the

original formula of CHARTET, formula 11, our demonstration is confirmed mathematically by this thesis that so long as Y_1 and H_1 are greater than zero, then

$$\left(\frac{Y_i}{H_i}\right)_{min} < \frac{\Sigma Y_i}{\Sigma H_i} < \left(\frac{Y_i}{H_i}\right)_{max}$$

It should be noted here that the value of $Y/H = k_1$ is more favourable than that with which we have worked so far, because in our example $k_1 = 2$ and Y/R did not reach this value until $R = \infty$. (See fig. 15.)

Having got the above result, let us return to the subject discussed in Chapter I. Let us take figure 1 again, but so that there still remains on the unloaded wheel a certain fraction of the original load. Q/4 (fig. 17). With the notations of Chapter I we can write once more the equilibrium equations

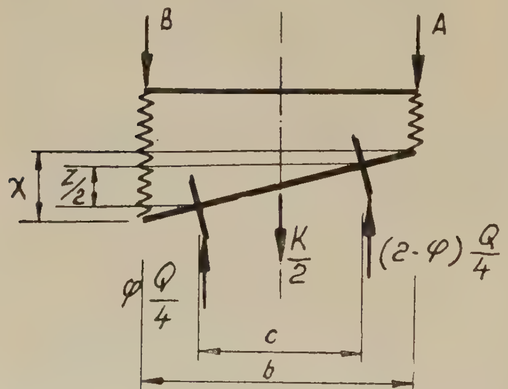


Fig. 17.

$$B = \frac{Q-K}{2} - A;\ \varphi \frac{Q}{4} \frac{b-c}{2}$$

$$+ (2-\varphi) \frac{Q}{4} \frac{b+c}{2} - \frac{K}{2} \frac{b}{2} - A.b = 0$$

and we will obtain with the unaltered constants of Chapter I as a final result that

$$Z_8 = f\left(\frac{c}{b}\right)^2 .Q.(1-\varphi) \qquad \text{(Formula 12).}$$

In example B of Chapter I for the standard gauge tank-wagon, we obtained Z_1=45 mm. With a flange having a degree of conicity equal to 70° according to CHARTET we get :

$$R_x = 0.35 \frac{Q}{4};\ \varphi = 0.35$$

i.e. there only remains 0.65 of Z_1 which can be counted upon, so $Z_8 = 29.3$ mm. It is true that on the other hand the height of flange must also be counted, for which a maximum conicity of 2° can be allowed, i.e. $ny = 8$ mm. Finally we get : 29.3 + 8 = 37.3 mm, a height upon which we can count. (See fig. 14*a*.)

We can therefore examine possible cases of derailment starting with the following principles drawn from what has been said above :

1) the load on the leading wheel should never fall below a value R_x;

2) there is no need to take into account such stresses as

$$\frac{Y}{H} \leq k_1,$$

because these only increase the safety factor against derailment or at least do not reduce it;

3) in the case in which

$$k' = \frac{Y'}{H'} > k_1$$

then it is necessary to determine a load R'_x which is greater than R_x, below which we cannot go owing to the danger of derailment.

This limiting value of the load can be found as follows :

$$Y' \leq k_1 (R'_x + H') - k_2 \frac{Q}{4},$$

whence

$$R'_x = R_x + Y' \frac{k' - k_1}{k_1 k'} \qquad \text{(Formula 13).}$$

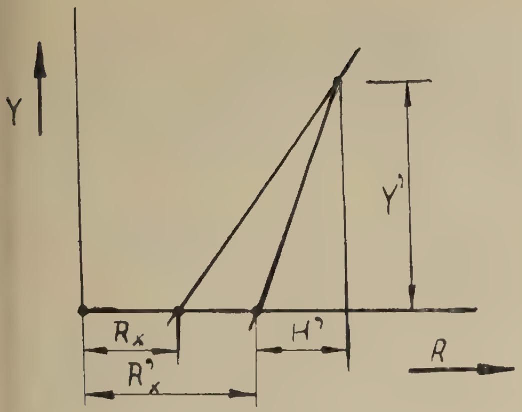


Fig. 18.

Let us now see in turn the chief transversal stresses which are generally taken into account. We do not intend to treat of each of them in detail, as detailed reports exist. We intend rather to mention them for the sake of completeness because they play an important part in the mechanics of derailment.

II. a.

Centrifugal force, when the radius of curvature is constant, means a transversal inertia force of constant value. It can be taken as in equilibrium with the superelevation of the rail. If the vehicle runs through the curve at a lower speed than the maximum speed corresponding to the superelevation, the resulting force tends to push the vehicle towards the centre of the curve. We can therefore leave this question which is outside our investigation.

II. b.

We can locate the centre of action of the wind upon standard gauge vehicles marked on figure 19 by the reading « s », at a height varying between 1 and 2 m. As the figure shows, the ratio between the horizontal load of the wheel flange and the additional vertical load will oscillate between the values :

$$\frac{Y}{H} = \frac{1.5}{s} = 1.5 \text{ to } 0.75.$$

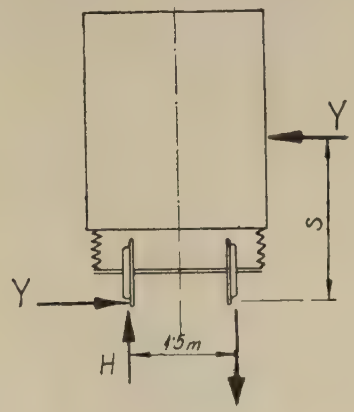


Fig. 19.

This shows that the force of the wind has no appreciable effect from the point of view of derailment on standard gauge vehicles; however a special examination is necessary in the case of very low vehicles, for example those with lowered frames. A similar result is obtained with narrow gauge vehicles.

(Naturally derailment must not be confused with the danger of overturning, which lies quite outside our subject.)

II. c.

Let us consider the thrust on the flange due to friction.

With the deviation of the vehicle on the rails, it is the friction on the running surface of the wheels which forms the resistance. If we wish to calculate the thrust on the flange needed to produce deviation, we must first of all determine the axis around which the vehicle turns. For this purpose HEUMANN's well known method has been used, based on the fact that nature tends to obtain its results with the *minimum* effort. (See *Bibliography* No. 13.)

In the following study it is sufficient to note that the friction of the two wheels of the leading axle has already been taken into consideration in Mr. CHARTET's theory. The friction of the rear axle is transmitted to the leading wheel by the guide plates and we can take it that this force lies above the rails at a height equal to the radius of the wheel.

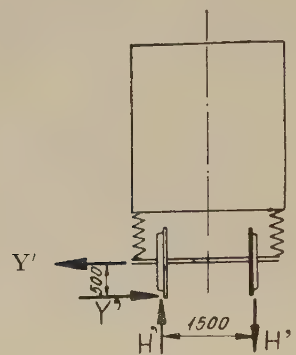


Fig. 20.

We can see from figure 20 that

$$\frac{Y'}{H'} = k' = 3 \text{ approx.}$$

This is an effect which must be given serious consideration. For example in the above case of a flange with a degree of conicity equal to 70°, where as we have seen $k_1 = 2$, we see that with a value of $k' = 3$, the minimum wheel load has a value of

$$R'_x = R_x + \frac{Y'}{6},$$

a value below which it is not possible to go. As the centre of friction is generally close to the centre of the rear axle, we can take as a suitable approximation, for the vehicle used in example I. *B*.

$$Y' = \frac{0.3 \times 1.5}{5} \frac{Q}{4} = 0.09 \frac{Q}{4}.$$

Instead of the original wheel load

$$R_x = 0.35 \frac{Q}{4},$$

in this case we can no longer allow a wheel load equal to $0.35 + 0.09$, i.e. $0.44 \frac{Q}{4}$ as the minimum. This further reduces the value of « Z » available as a reserve from the derailment point of view.

Naturally, in this simplified report, various simplified hypothesis have been allowed. Thus instead of the flanges, it has been allowed that it is the running surface which is borne upon; in addition, the loads on the wheels of a mounted axle are taken as equal. The two wheels of a mounted axle are taken as being cylindrical surfaces of identical radius, and finally it has been taken that the coefficient of friction is the same for all the wheels. On this latter point, LABRIJN (see *Bibliography* No. 14) finds that the coefficient depends to a large extent on the relation between the transversal displacement and the forward speed, i.e. in the case of a small angle of attack, it depends upon the distance the wheel is from the centre of friction (see *Bibliography* No. 15). We have seen in our report that in order to be derailed, the leading wheel must be to some extent freed of its load. If this occurs, then in general the wheel diagonally opposite the leading wheel will also be freed of its load and it will consequently be necessary to revise the definition of the centre of friction.

Let us therefore consider a four wheeled vehicle and see how the axis of rotation will be displaced if on the rear mounted axle the load on one wheel is progressively transferred to the other wheel. Naturally, here again, it is necessary to simplify things the better to appreciate the result, as we have already done when discussing the problems of the railway (see fig. 21).

We will therefore retain all the simplifications indicated above with the exception of that concerning the loads on the rear wheels. These latter differ and their ratio B/A or what is the same thing B*f*/A*f* may

vary between zero and one. In order to simplify once more, we have taken it, as already stated above, that the reactions of the leading mounted axle are vertical and applied on the axis of the middle of the rails and that they lie within CHARTET's formula.

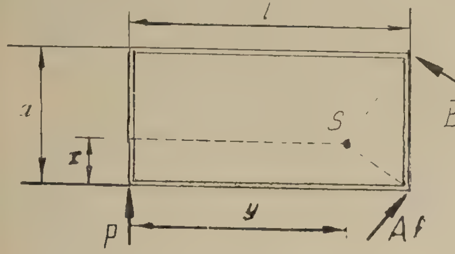


Fig. 21.

S = axis of rotation of friction.

Figure 21 represents our vehicle in plan. S is the centre of friction. The equilibrium of the stresses gives us the equation :

$$Py = Af\,[x^2 + (l - y)^2]^{\frac{1}{2}} + Bf\,[(a - x)^2 + (l - y)^2]^{\frac{1}{2}}$$

which can be written :

$$\frac{1}{f}\frac{P}{A} = \frac{l}{y}\sqrt{\left(\frac{x}{a}\right)^2\left(\frac{a}{l}\right)^2 + \left(1 - \frac{y}{l}\right)^2} + \frac{B}{A}\sqrt{\left(1 - \frac{x}{a}\right)^2\left(\frac{a}{l}\right)^2 + \left(1 - \frac{y}{l}\right)^2}.$$

Here, we have to find the minimum value of P/A because we know that this minimum determines the centre of friction. We have four parameters a/l, y/l, x/a and B/A. Let us take the first, a/l equal to 0.5. We can then represent the values of P/A for all values between zero and one of the three others, i.e. y/l, x/a and B/A. This is shown in figure 22. Above the vehicle, in line with each point « S », we have shown in the vertical direction the value P/A which is necessary to cause deviation when the centre of rotation of friction lies exactly on « S ». This is an axonometrical figure. There are separate surface curves corresponding to various values of the parameter B/A. Naturally, these values of P/A do not exist in reality; we have only shown them as minimum values in order to make it clearer how the value is to be found.

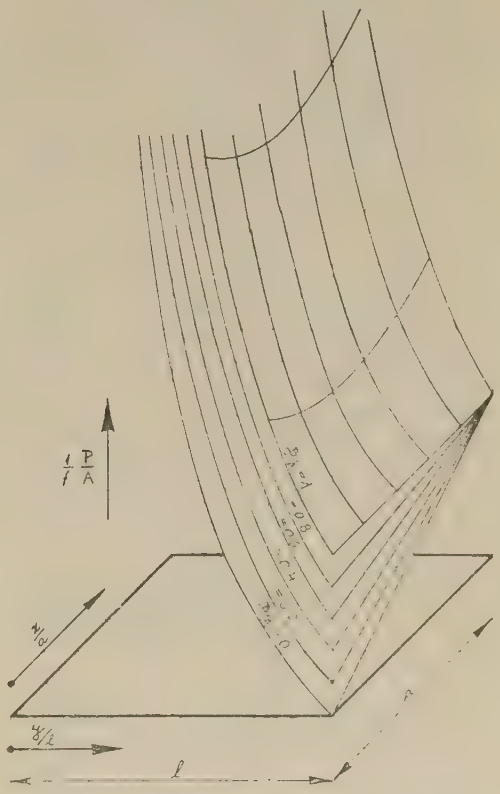


Fig. 22.

In the case B/A = 1, the highest surface is that which is valid. We have traced the line of intersection of the surface curve for certain constant values of y/l. The minimum values in reality fall within the axis of the vehicle, which agrees with the hypothesis of the minimum method of HEUMANN. The lower part of this surface lies above the rear axle A — B, where the generatrix becomes deformed and flattens out. Consequently above the rear axle, we get the

minimum value at each point. In what follows we shall see that for various reasons the axis of rotation lies in reality a little nearer the first axle and consequently if $B/A = 1$, the minimum value falls precisely on the axis of the vehicle.

As soon as the ratio B/A decreases, the phenomenon being approximately symmetrical compared with the axis of the vehicle and having a downwards bulge, there is a weakening and the minimum value moves frankly towards point « A ». We can see therefore that if the load is taken off, the centre of friction comes nearer and nearer to the wheel behind the leading wheel.

II. d.

If the vehicle attacks the outer rail at an angle τ, the force of deviation (perpendicular to the track) has to overcome not only the transversal friction but also the inertia forces. Let us suppose that the vehicle is moving at a speed of V m/sec. According to figure 23, the mounted axle will then have to have a speed equal to $V_0 = V \sin \tau$ in order to be able to follow

Fig. 23.

the rail. The kinetic energy needed to obtain this result is equal to

$$\frac{1}{2} M_r V_0^2 = \frac{1}{2} M_r V^2 \sin^2 \tau.$$

Owing to the shock, the rail moves horizontally. In addition, the wheel, the axle, bogie, etc., and all the parts are acting as spring, and we can express what happens by considering that the mass collected at the centre of shock M_r strikes through a spring whose constant is c kg/m against the infinite mass of the track. If the maximum displacement is s mm, and if we designate the maximum force of the shock by Y, then the work done during the shock is equal to $sY\frac{1}{2}$.

Then

$$\frac{sY}{2} = \frac{M_r V^2}{2} \sin^2 \tau$$

and, as $s\,c = Y$, we get finally :

$$Y = V \sin \tau \sqrt{M_r c} \qquad \text{(Formula 14)}$$

(see *Bibliography* No. 15). This maximum lateral force « Y » makes the mass of the vehicle revolve around the axis passing through its centre of gravity. In addition, it displaces the whole of the vehicle as well in the manner shown in figure 24.

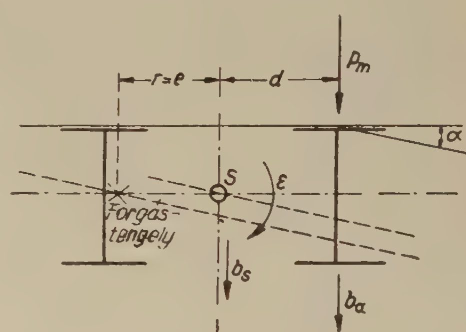


Fig. 24.

Forgâstengely = axis of rotation.

Adding these two effects together, we can take it that the vehicle turns, not around « S », the axis passing through the centre of gravity, but around an axis lying farther back. It then becomes necessary to consider the reduced mass in relation to this axis. Once again, we will not go into details, because these can be found in abundance in the bibliography relating to this question. We must however remark upon several essential points of view as we did when discussing friction.

Let us quote for example LABRIJN, who

has endeavoured in the very simple manner described above to determine the maximum speed allowable (see *Bibliography* No. 6). For example, he supposes that an axle load of 12 t and a force of shock Y=6 000 kg takes place in the lateral direction. The formula he gives enables a vehicle to take the rail at an angle of attack and at a speed resulting from the equation

$$\sin \tau = 2.235\sqrt{\frac{s}{V}}.$$

This equation is represented diagrammatically in figure 25.

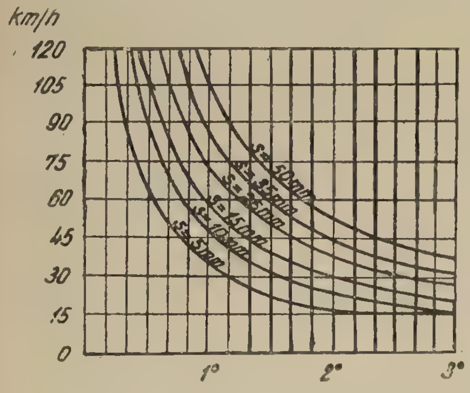


Fig. 25.

According to experience obtained to date, an angle of attack of 2° is permissible up to a speed of 45 km (28 miles/h). According to the diagram this gives a total deformation of the springs $s = 40$ mm, and, on this basis, at a speed of 120 km (74 miles/h), an angle of attack of about 50' would be admissible.

In this way, the problem appears to be simplified excessively. Thus LABRIJN took in his estimate a maximum value Y/R=1; on the other hand, he did not take into consideration that in this force Y the friction is already included and finally, how can this lateral displacement of 40 mm conform to reality? In particular with the data taken by LABRIJN (Y = 6 000 kg, $s = 40$ mm, $1/c = 40/6\,000 = 6.7$ mm/t), the constant of the spring which was unknown is determined. It is true that we have not yet got any experimental data regarding this elasticity. However, according to calculations (see *Bibliography* No. 16), the flexibility of the rails, the mounted axles and the frame added together cannot exceed approx. 1.34 mm/t. Starting from this, only a very low speed can be allowed for the usual angles of attack, since according to the above figures of LABRIJN, the value of « s » is just one fifth of that we expected. How does this contradiction arise?

In our opinion, to reduce the whole mass of the vehicle at the point of shock of the flange of the leading wheel is a supposition which does not agree with reality. Fundamentally, we have a track and an axle mounted relatively rigidly, i.e. with low flexibility per kg. It is through this relatively unelastic medium that the running parts have to be impelled, the mass of which compared with that of the whole vehicle is small, about one tenth. The other parts of the vehicle take part in the *transversal* shocks through much softer springs. For example the U.I.C. in notice B 35 has fixed the measure of transversal elasticity of the axle guards at about 7.4 mm/t including the torsion of the frame. On this basis, let us continue to examine the shock of this mass, or rather this oscillating mass. By simplifying we can take it that we have to deal with a system of three masses.

According to figure 26, one of the three masses is the earth which we will consider

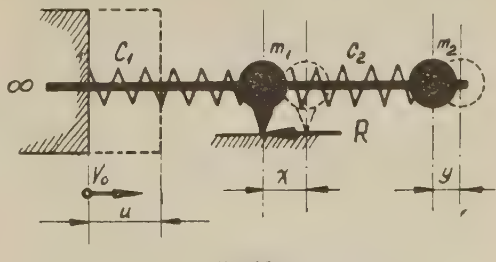


Fig. 26.

as infinite. The second is the mounted axle plus the masses of the rolling gear (axle boxes, without springs). It is the leading driving axle which is in question.

This second mass will be designated by m_1. The third is that of the frame and body transferred to the point of contact of the flange of the leading mounted axle. We will designate this by m_2. Between the infinite mass and m_1, there is a spring of flexibility c_1 (kg/m), which represents the whole of the transversal elasticity of the mounted axle and of the track, whereas between m_1 and m_2 there is c_2 (kg/m), i.e. the flexibility of the frame and the guard plate.

A transversal friction also acts upon the mass m_1. This is the whole transversal friction of the front mounted axle. Let us look for the maximum stress in the spring c_1 when the masses m_1 and m_2 strike through the two springs against the infinite mass at the same speed V_0. To simplify the calculations, we will transpose things and take it that at the beginning m_1 and m_2 are at rest, whereas the infinite mass is approaching them at a speed V_0. The relative displacements of the masses are not changed and consequently we have an image closer to reality.

If the force occurring in the first spring is P_1 and that in the second P_2, we can write :

$$u = V_0 t; \quad P_1 = c_1 (V_0 t - x); \quad P_2 = c_2 (x - y);$$
$$P_1 - P_2 - R - m_1 x'' = 0; \quad P_2 - m_2 y'' = 0;$$

and we find for y and x a non-homogenous differential equation of the fourth degree and of identical form (y^{IV} representing the fourth derivative of y).

$$m_1 m_2 y^{IV} + (m_1 c_2 + m_2 c_1 + m_2 c_2)\, y'' + c_1 c_2 y = c_1 c_2 V_0 t - c_2 R$$
$$m_1 m_2 x^{IV} + (m_1 c_2 + m_2 c_1 + m_2 c_2)\, x'' + c_1 c_2 x = c_1 c_2 V_0 t - c_2 R$$

The roots of the characteristic equations relating both to x and y have as their value :

$$\alpha_1 = i\sqrt{\frac{(m_1 c_2 + m_2 c_1 + m_2 c_2) + \sqrt{(m_1 c_2 + m_2 c_1 + m_2 c_2)^2 - 4\, m_1 m_2 c_1 c_2}}{2\, m_1 m_2}}$$

which we can designate briefly as $\alpha_1 = i\sqrt{+}$

and $$\alpha_2 = i\sqrt{\frac{(m_1 c_2 + m_2 c_1 + m_2 c_2) - \sqrt{(m_1 c_2 + m_2 c_1 + m_2 c_2)^2 - 4\, m_1 m_2 c_1 c_2}}{2\, m_1 m_2}}$$

which we can designate in a similar way by $\alpha_2 = i\sqrt{-}$

The solution of the differential equations, which present the oscillations, is the sum of the two sinus functions. It is given by the following functions :

$$x = A \sin(i\alpha_1 t + \varphi) + B \sin(i\alpha_2 t + \psi) + V_0 t - \frac{R}{c_1},$$

and

$$y = K_1 A \sin(i\alpha_1 t + \varphi) + K_2 B \sin(i\alpha_2 t + \psi) + V_0 t - \frac{R}{c_1}.$$

In this latter

$$K_1 = \frac{m_1}{c_2}\alpha_1^2 + \frac{c_1 + c_2}{c_2};$$

$$K_2 = \frac{m_1}{c_2}\alpha_2^2 + \frac{c_1 + c_2}{c_2}.$$

Now we can determine the initial conditions. It will be seen from figure 26 that as long as the displacement « u » of the infinite mass is not such that the force of the spring resulting exceeds the friction R, the masses m_1 and m_2 remain at rest. Up to this point, the speed is naturally nil. Consequently, since

$$V_0 t = \frac{R}{c_1},$$

we get as the initial conditions :

$$x = 0; \; x' = 0; \; y = 0; \; y' = 0.$$

From this we conclude :

$$\varphi = -i\alpha_1 \frac{R}{c_1 V_0} \text{ and } \psi = -i\alpha_2 \frac{R}{c_1 V_0},$$

i.e. the two sinusoidal oscillations both start from zero after a period $t_0 = \frac{R}{c_1 V_0}$. For this reason and to simplify matters, we will in future represent the oscillation by taking as the origin of the period the moment t_0. We can then continue our calculations by taking $\varphi = \psi = 0$. And the values of the amplitudes of the oscillations are

$$A = \frac{V_0}{\sqrt{+}} \frac{K_2 - 1}{K_2 - K_1}$$

and

$$B = \frac{V_0}{\sqrt{-}} \frac{1 - K_1}{K_2 \quad K_1}.$$

The oscillations are represented by the curves of figure 27.

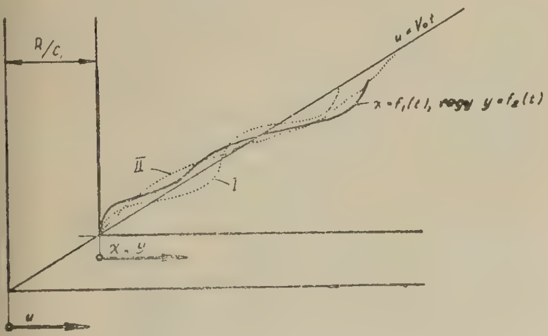


Fig. 27.

The Hungarian term vagy = or.

The oblique line $u = V_0 t$ marks the displacement of the infinite mass; the dotted lines I and II represent the component oscillations and the sinuous line shows the oscillations of the masses m_1 and m_2. It is therefore necessary to measure the displacement of the infinite mass (u) from the vertical axis of the coordinates up to the oblique line, whereas the displacements x and y of the masses m_1 and m_2 have to be measured horizontally from the vertical line drawn at a distance R/c_1 from the origin. As the force developing in the spring is proportional to the difference between the two distances, the force developed in the spring « c_1 » at a given moment t will be given by the distance measured horizontally between the line representing the oscillation and the oblique line. This distance must be multiplied by the constant c_1 of the spring to obtain the force. To this must be added the resistance R, but this has a constant value and only concerns us from the point of view of the solidity of the vehicle and the running parts. We will not go into this point here, and the friction of the first mounted axle is included in the study on derailment by CHARTET; for this reason we will not bother about this force R in what follows. In addition, we will simplify our figure as stated above by only beginning our examination after the moment t_0; we will then only consider the axes of the coordinates displaced to the right and upwards.

Of the oscillation we have only to retain the half of the period (π) because the two springs can only transmit pressure, since the masses are not connected to the springs. The portion of the period during which the spring works in traction does not exist in reality.

We can make another important simplification in our figure. In all the terms of the formula giving x, the speed V_0 intervenes. We can divide the whole and then, according to the formula $x/V_0 = f(t)$, our figure will give the force of the spring for all the corresponding speeds, if we multiply the horizontal distances given above by $c_1 V_0$.

The oscillation represented by the line I is faster than the other and relates essentially to the shock of the mounted axle the mass of which is relatively small and the elasticity lower, whereas line II shows a longer oscillation indicating the predominant part of the vehicle, which is larger in size and has softer springs.

Let us now see by means of two examples how the situation can be expressed in

numerical values. Let us take it that the total weight of the two rolling parts is :

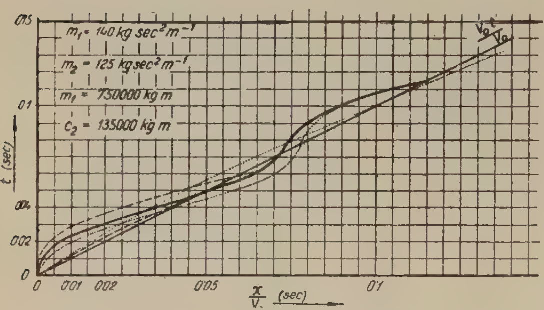
$$K = 2\,750 \text{ kg};$$

$$c_1 = 750\,000 \text{ kg/m};$$

$$c_2 = 135\,000 \text{ kg/m}.$$

From their amounts, these appear to be close to actual values. Let us consider a vehicle with a small tare, Q = 7 680 kg and a vehicle with a 16 t axle load, i.e. 32 000 kg total weight.

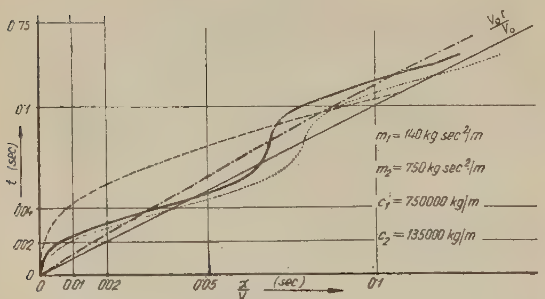
A. In the case of the light vehicle m_1



——— Oscillating masses m_1 and m_2 controlled by the springs c_1 and c_2.

– – – – Mass $(m_1 + m_2)$ controlled only by the spring c_1 or $c_ö = \frac{c_1 c_2}{c_1 + c_2}$ (The latter with abscissae reduced in the ratio $c_ö/c_1$.)

Fig. 28.



——— Oscillating masses m_1 and m_2 controlled by the springs c_1 and c_2.

–·–·– Mass $(m_1 + m_2)$ controlled only by the spring c_1 or $c_ö = \frac{c_1 c_2}{c_1 + c_2}$ (The latter with abscissae reduced in the ratio $c_ö/c_1$.)

Fig. 29.

= 140 kg/sec²/m and taking it that the axis of rotation is close to the rear axle, $m_2 \simeq 125$ kg/sec²/m. With these values we get the ratio :

$$\frac{x}{V_0} = t - 0.01 \sin 80.3\, t - 0.066 \sin 29.9\, t$$

which we have represented in figure 28.

B. In figure 29, we have a curve corresponding to values $m_1 = 140$ kg/sec²/m and $m_2 = 750$ kg/sec²/m, using the whole axle load. This gives us

$$\frac{x}{V_0} = t - 0.0106 \sin 79.3\, t$$
$$- 0.0131 \sin 12.15\, t.$$

By the simple method of calculation of LABRIJN or DAUNER-HILLER, reducing the whole to a single mass, we obtain the following form :

$$\frac{x}{V_0} = t - \sqrt{\frac{m}{c}} \sin \sqrt{\frac{c}{m}}\, t$$

The question now arises of knowing if in the calculation of the simple shock of the total mass $(m_1 + m_2)$, it is necessary to take into consideration only the spring c_1 or to consider the arrangement shown in figure 30 in which the combined constant is

$$c_ö = \frac{c_1 c_2}{c_1 + c_2}$$

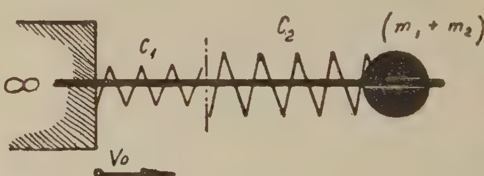


Fig. 30.

In figures 28 and 29, we have shown by broken lines the quarter period of the two combinations of the oscillating masses. Naturally, in the case of $c_ö$ it was necessary to reduce the values of x/V_0 in the ratio of $c_ö/c_1$, as in reality it is the forces of the springs in which we are interested and it

is only in this way that the data can be compared one with the other. Naturally, if we calculate using c_1, we will have the greatest amplitude of the oscillations, and with c_0 the smallest.

According to figure 28, it can be seen that in comparison with the more accurate calculation, the simplified calculation gives a maximum force about 30 % higher for spring c_1, whereas calculating with spring c_0, we get only 20 %.

In the same way, according to figure 29, the calculation with the spring c_1 gives 150 % more than the more accurate calculation, whereas calculating with c_0, we get about the same value.

All this enables us to take it as proved that this simple method of calculating cannot be used to determine the forces of shock in the horizontal plane, if there are also springs acting on the horizontal plane between the vehicle and its running gear.

There is also a very interesting fact to be noticed on figures 28 and 29. The first quarter periods of the line representing the oscillation are almost completely identical. This shows that in the case of vehicles with sufficiently flexible axle guards whether the vehicle we are considering be loaded or empty, the force of the lateral shock (leaving aside the transversal friction of the vehicle) has practically the same value and undergoes the same variations. In other words, the force of the shock which reaches the vehicle depends for the greater part upon the mounted axle and the running gear; the body and the load only have a very slight influence. This thesis is still to be proved by experiments.

It should be noted here that DAUNER-HILLER in the article already quoted several times, drew attention to the need for such experiments, but we do not know if he has published the results of any such experiments.

This being so, let us examine the disposition of the forces of shock in space. In the case of the shock to the mounted axle and running gear, the forces occur essentially as shown in figure 20. We have already pointed out that this is an effect which should be taken seriously into consideration from the point of view of derailment, owing to the value $k' = 3$ approx. On the other hand, if we consider the shock of the whole of the masses of the frame and body, the forces occur as shown in figure 19. We can estimate the height of the centre of gravity « S » of the vehicles as between 1 and 2 m; therefore the value of the ratio $Y'/H' = k'$ will lie between 1.50 and 0.75.

Consequently, here again we arrive at this result that from the point of view of danger of derailment, in general in estimating the lateral shock the masses of the frame and body can be neglected. Since it is actually speaking only the first quarter of the period of oscillation which interests us from the point of view of our examination — as it is during this first quarter that the greatest stress occurs — we can also say that during this part of the period the mounted axle and the suspension gear take part in the shock it might be said independently of the other masses of the vehicle. (This is made possible also if, at the moment of shock, the box has not yet come in contact with the axle guard, i.e. if the mounted axle is acted upon in its middle position). Consequently, following the shock, when the forces situated higher up come into action, the load on the leading wheel will increase to such an extent that the danger of derailment will be diminished. Naturally, a special examination will be required in the case of vehicles with a low centre of gravity to which the above generalisation does not apply.

We have seen that on the one hand the masses apart from the mounted axle and the running gear with its own springs have only a small influence on the force of the shock, but that on the other hand these masses need not be taken into account owing to the increase in the load on the wheel. Thus, we might say, on the basis of this theroretical study, that for the lateral shock it is sufficient from the point of view of derailment to count only the

shocks of the mounted axle and the suspension gear, i.e. we can return to the simple method of DAUNER-HILLER. But we now know that instead of $m_1 + m_2$, we need only take account of the mass m_1 and the spring c_1.

In order to avoid misunderstandings, we must make ourselves quite clear and insist that our facts are only valid as regards derailment, whereas for solidity, etc., the more exact method must be used. According to our formula 14, the value of the lateral shock is :

$$Y' = V \sin \tau \sqrt{m_1 c_1}.$$

In addition, we know that $k' = 3$ and with this value, according to what has been said in chapter II. *c*.

$$R'_x = R_x + \frac{1}{6} Y' = 0.35 \frac{Q}{4} + \frac{1}{6} Y'.$$

Finally we also know from what was stated at the end of chapter II, *b*, that with a load on the wheel of $\varphi Q/4$, we must always keep above R_x', i.e. it is necessary to have

$$\varphi \frac{Q}{4} \geqslant R'_x = 0.35 \frac{Q}{4} + \frac{1}{6} \sqrt{m_1 c_1} \sin \tau$$

or more simply

$$\varphi \geqslant 0.35 + \frac{2}{3} \frac{Y'}{Q} \qquad \text{(Formula 15).}$$

It is true that we have linked up this result with a degree of conicity of the flange equal to 70°, but as it is the order of magnitude with which we are mainly concerned, we will continue to count on this value in what follows. In figure 31 we have traced for a light vehicle with a tare $Q = 7\,680$ kg and for a loaded vehicle whose weight is $Q = 32\,000$ kg, the values of φ in hundredths of τ for various values of V and τ. (Owing to the smallness of τ, we have been able to represent the values of φ by straight lines). If we start with the values admitted by LABRIJN in the example he dealt with, where a speed of 45 km (28 miles)/h is allowed up to an angle of 2°, whereas according to the figure we get a minimum value for the wheel load $\varphi = 74\%$ approx. Let us take it that the complement i.e. 26 % is sufficient to compensate the faults in construction and poor maintenance (of the vehicle and the track). Admitting the constancy of these factors, we find in tracing the horizontal towards the left, that at a speed of 120 km (74 miles)/h, it is possible to admit an angle of approx. 45′ at the most. The result is nearly the same as that found by LABRIJN in his afore-quoted study.

It may be asked with reason why it was necessary to proceed with this long discussion on the loads of the wheels and on the theory of CHARTET and the study of oscillations whereas by following the method of LABRIJN we arrived at the same result in a much simpler fashion. But it must not be forgotten that LABRIJN worked on data selected completely arbitrarily; we might even say that he selected his data in order to obtain a result which would conform with railway experience. But it is not our purpose to demonstrate the falsity of railway experiments carried out over long years (which would moreover be in vain), but as we have already explained above, our object is to supply the theoretical foundation for the results of experiments, so as to smooth the path for future progress.

However figure 31 must not be taken as a result to be used in the future. We have left out a great many factors which are without doubt essential. For example, on lines generally run over at low speeds, greater tolerances are allowed as regards maintenance. In the same way, greater wear is allowed in the case of vehicles for low speeds than in the case of vehicles for fast speeds. At higher speeds, there are the greater dynamic effects, etc., to be taken into account. These are factors which must be taken into account, but do not let us anticipate our next chapter in which we shall deal with this point in greater detail.

It should also be noted that during our above report, we have supposed that we

know the centre of rotation around which the vehicle turns. It is true that we can determine the centre of friction by HEUMANN's process of the minimum, but we have seen that when a wheel is freed of its load the centre varies from the axis of the vehicle. When determining the effect of the forces of inertia, it is necessary to count upon rotation around a different

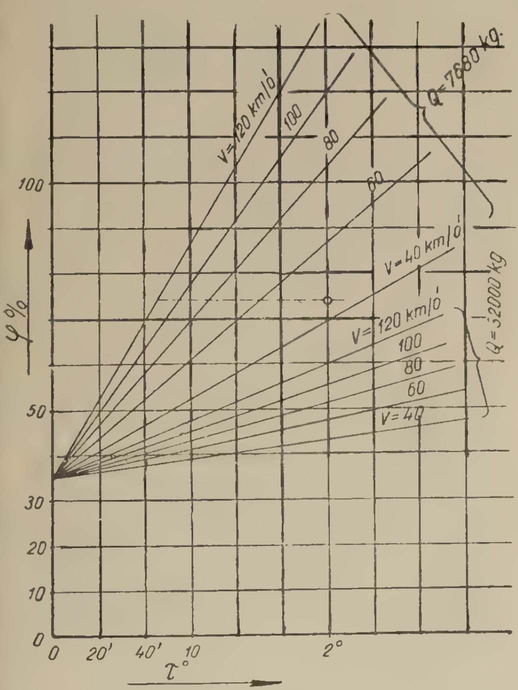


Fig. 31.

N. B. — km/ó = km/h.

axis. It is therefore natural to admit that the vehicle turns around a centre lying between the two. DAUNER-HILLER recommends (see *Bibliography* No. 15) that the force of friction be determined separately in considering the rotation around the centre of friction, and the effect of the masses in considering a rotation around the centre determined by the forces of inertia. The two can then be added together. In conclusion, we may state that most of the questions which have not been solved as regards derailment relate precisely to the real intensity of the shocks between rail and wheel. Here again extensive experiments are lacking for which the theory adopted would serve as a base. The extraordinary difficulties encountered with such experiments explain why no thorough investigation has so far been carried out. Measurements have already been taken to ascertain the reaction of the flange on striking certain parts of the rail. This can only be done at certain points of the rail, which is why we do not know if the maximum force was measured or only a fraction of this maximum. Measurements have also been taken on locomotives (see *Bibliography* No. 3). The reactions between the mounted axle and the box were measured by means of the electric resistance of carbon plates. As we saw above, the determination of the horizontal forces, precisely the most important effect, that of the masses of the mounted axle, cannot be measured and that is why they must be determined by calculation. It would be necessary to pick up by means of some precision instrument the reaction occurring between the rail and the flange while the vehicle is in movement. Perhaps, this will be done by means of an instrument on the lines shown in figure 32, using the elastic deformation of the axle. The instrument should record continuously the deformations at a given point on the whole circumference of the wheel.

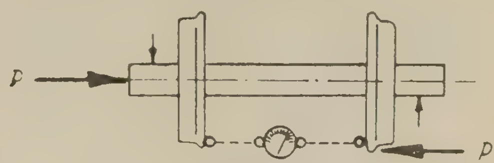


Fig. 32.

On traction vehicles to the lateral effects must be added the horizontal component of the traction effort, the effect of which in general is to increase the safety. In addition to this, account must be taken of the forces and friction occurring in the shock gear, as well as the forces caused by transversal oscillations.

CHAPTER III.

We will end by summing up our study. The essential fact is that with each vehicle there is a barrier which must not be passed, a height $Z + ny$, which should prevent the vehicle from being derailed.

The value of « Z » depends on the constructional details of the vehicle. These details are :

I. *a)* the ratio between the gauge of the track and the distance apart of the springs;

b) the flexibility of the springs;

c) the coefficient of torsion of the frame and body;

d) the tare of the vehicle.

The value of « *ny* » depends on the one hand :

II. on the constructional details of the vehicle :

e) the profile of the tyre;

f) distance between centres of the axles;

g) diameter of the wheel;

and on the other hand :

III. on the constructional details of the track :

h) the angle of attack;

i) the rail profile.

Of the theoretical *limiting height* determined in this way, some part is absorbed by the following factors :

IV. Construction of the vehicle :

a) torsion of the frame and body (deformation);

b) inaccuracies in manufacture;

c) friction in the springs and in the running gear;

V. Operating and maintenance of the vehicle :

d) asymmetrical loading and unloading (for example the unloading of the water tanks and coal bunkers, etc.);

e) friction (increasing in the springs and running gear, compared with their condition on manufacture);

f) unequal wear of the suspension gear;

g) broken springs, etc.

VI. As regards the track :

h) constructional standards (superelevation gradients);

i) differences in the dimensions on construction (permissible tolerances).

VII. Alterations in the condition of the track (maximum drop allowed in the rails between two periodic overhauls).

VIII. Transversal stresses (the heights needed to produce the forces R_x, R_x').

IX. Dynamic vertical oscillations due to the movement of the vehicle.

Now, having gone into the question of derailment at length, an engineer will at once ask :

Would it not be possible to divide up the height as defined and distribute it according to the points summed up above, into zones in such a way that certain points had their allocation marked in these zones? Perhaps, it would be possible to apply in this case also the usual method of tolerances in manufacture.

For example, if we measure the load on each of the four wheels of a vehicle suitably balanced, and if we estimate the influence of these loads, then we can take away from « Z » the value corresponding to points I and IV and see what remains of Z. Knowing the destination of the vehicle, we could add to this the value of the « *ny* » depending upon points II and III, and again substract the height needed for points VI and VIII. From what remains of the limiting height after all this, we should take away a zone for the point.

V. Which should be respected by the operating department which maintains the vehicles, and likewise a zone for the point

VII. which should be respected by the permanent way maintenance department.

Finally, there remains to verify if the remaining zone of the limiting height is sufficient to cover the extent of the dynamic oscillations of point IX, which could be done by making a trial run.

Naturally, the question of the margin of safety arises in this case which would involve having an additional supplementary zone. Perhaps it would be sufficient for this *margin of safety* if all the unfavourable factors mentioned did not occur at one and the same time and at the same place. Such a margin exists for example between the loading gauge and gauge of the permanent way. Operating experience must deal with this question of the margin of safety. They should determine, by reconstituting the data examined here, if there remains a zone of safety for the vehicles in service. In our opinion, those questions the solution of which depends upon our own practical good sense can only be judged technically upon such a basis. For example, to decide if a vehicle running on a certain line is not likely to be derailed, and up to what speed this is so, we have to depend completely upon our own judgment. (It may often happen that the uneventful running of the vehicle corresponds to the maximum speed without revealing where the danger of derailment lies. There are two different points of view which should be examined separately.) In this field, we are at the present far from the ideal task of the engineer who designs a mechanism and is able to fix the constructional tolerances, etc., thanks to which the vehicle will behave as designed to do.

The fixing of certain amounts in the zones should be undertaken by the International Railway Union (for example the permitted tolerances for the track, the superelevation gradient, dropped rails, etc.). The result would be, for example, that in the case of fast vehicles the minimum tare would no longer be laid down but according to what has just been said a safety zone of a certain width specified.

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gives needed clearance in St. Clair tunnel.

(From Railway Engineering and Maintenance, February, 1951*).*

The Canadian National has for many years been detouring high box cars and open cars with high loads around its St. Clair tunnel under the St. Clair river between Sarnia, Ont., and Port Huron, Mich. The detouring of these cars became necessary because the original track structure was such as to limit the vertical clearance to 16.33 ft. above the top of rail. This clearance was further reduced to 15.33 ft. by overhead trolley wires installed for a traction system. How the road was able to gain 6 in. additional clearance and thus end the detour is told herein.

Through the use of a type of track construction including a continuous concrete stringer under each rail, carrying an arrangement of continuous steel plates and malleable iron chairs for supporting the rails, the Canadian National was able to obtain additional clearance in its important St. Clair tunnel, immediately west of Sarnia, Ont. This tunnel connects the Canadian National with its American subsidiary, the Grand Trunk Western. The tunnel passes beneath the St. Clair river between Sarnia and Port Huron, Mich. It is of great importance to the C. N. R. because it permits direct movement of traffic between Chicago and eastern Canada.

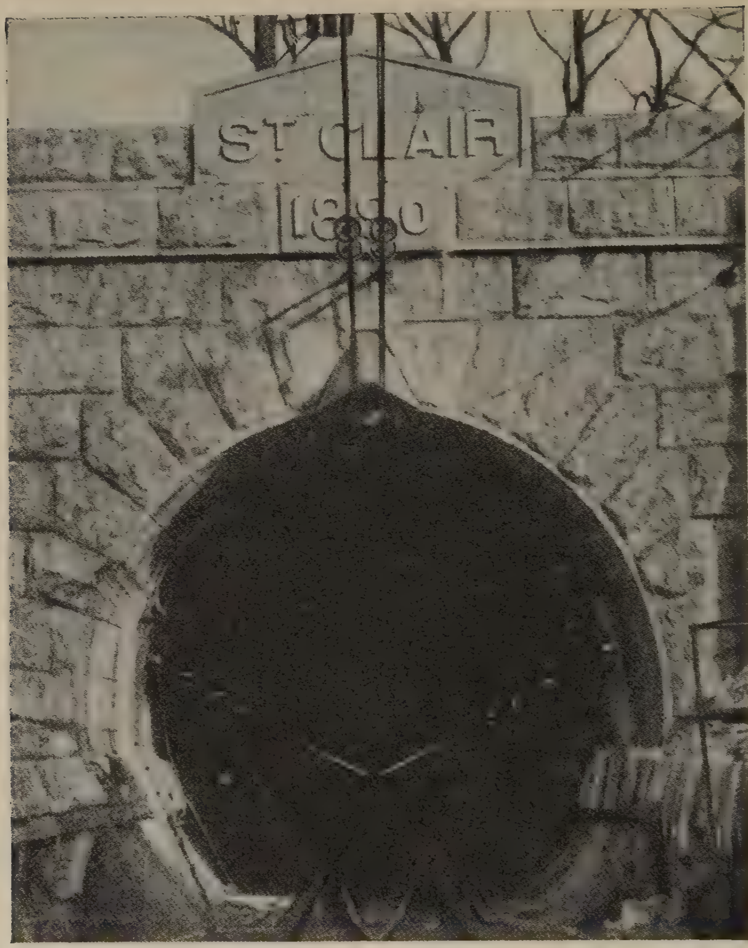
The clearance difficulties were due to the fact that in the construction of the tunnel the vertical clearance was restricted to 16.33 ft. This was later reduced to 15.33 ft., when an overhead electric traction system was installed. Thus, as the height of box cars began to increase, a point was eventually reached where all cars of a height greater than 15 ft. above the top of rail had to be detoured around the tunnel.

When eastward trains of the Grand Trunk passed through the yards at Battle Creek, Mich., the high cars were cut out and placed in trains for Detroit, where they were ferried across the Detroit river to Windsor, Ont. Detouring of westbound cars began at the yards in London, Ont., and was conducted in the same manner to Battle Creek. The cost of the detour operation is estimated as amounting to $ 15 per car. The number of cars so handled in a year's time represented a substantial part of the operating expenses of the two districts involved. Further, a delay of at least a halfday per car resulted from the rerouting, and for this reason it is possible that many shippers assigned their excess-dimension cars to other roads, resulting in a direct loss of revenue to the C. N. R.

The St. Clair tunnel was built in 1889-1891. It is a single-track, shield-driven bore, constructed of a number of flanged segments of cast iron. These segments are bolted together to form rings 1 ft. 6 1/4 in. wide and with inside and outside diameters of 19 ft. 10 in. and 21 ft., respectively. The rings, in turn, are bolted together circumferentially to form the tunnel proper which is 6.028 ft. long. Of this length. 2 699 ft. are in the United States and the remainder, 3 359 ft., is in Canada.

The approaches to each end of the tunnel are on descending grades of 2 per cent, which continue for 2 428 ft. into the tunnel on the American side and for 1 900 ft. on the Canadian side. For a distance of about 1 700 ft. in the mid-section of the tunnel the grade is practically level.

When the tunnel was built, its bottom was filled with concrete, brick and mortar to depths varying between 8 in. and 20 in., to form the foundation for the track struc-



Shape and construction of St. Clair tunnel bore may be appreciated by this view of the American portal.

ture. Four rows of continuous timber stringers were then laid directly on the concrete, and crossties were laid over the stringers, being secured with drift bolts. The rails, supported on tie plates, were secured to the crossties.

Maintenance difficulties.

This arrangement left no room for adjustment and after the electrification, in 1908-09, the clearance was fixed at 15.33 ft. While this figure may have been acceptable at that time, the continued growth of box cars and the retirement of older cars gradually led to the serious condition which resulted in the project to get increased clearance.

In addition to the clearance problem, the work of maintaining the track has been difficult in some respects. For example,

the wood stringers and the concrete suffered heavily from mechanical wear and the use of shims was resorted to at many points. Anchorage of the rails was also a problem because of the sharp descending grades at each end of the tunnel.

A review of the problem indicated that the most practical way to increase the clearance would be to remove the existing tie-and-stringer construction and replace it with a new type of structure, involving only two stringers, these to be of concrete anchored to the old concrete foundation. The plan also called for continuous steel cover plates, anchored to the concrete stringers and supporting malleable iron chairs for holding the track rails in proper position. In this way the clearance could be increased 6 in.

Showing the forms and reinforcing in place for the new concrete stringers. The old timber stringers were shifted so that the new ones could be properly positioned.

Daily occupancy — 2 ½ Hr.

The initial phase of the project — that of preparing the foundation — was begun in the early fall of 1948. Because of heavy traffic in the tunnel, it was necessary to set up definite arrangements for the construction forces to have the use of the track. This resulted in a daily period of track occupancy, beginning at 1.30 p.m. and ending at 4.00 p.m.—2 1/2 hr. On a few occasions, this was extended until as late as 5.00 p.m.

This 2 1/2-hr. period made it necessary to plan each day's work with care to assure

that steady progress was made. In many cases, where the track was not obstructed, work could be progressed outside of the track-occupancy period, using flag protection as a warning system.

In the first phase of the project, which involved the preparation for the concrete stringers, all of the old track was removed and the old wooden stringers were shifted to clear the location for the new concrete members. The old concrete foundation was inspected and all defective material was removed and the ties and rails were replaced. Forms for the new stringers, and steel reinforcing, were placed in position under traffic. The final step, before concreting, was to wash the surface of the old concrete within the forms until clean, after which it was sprinkled with Ferritex powder, a material produced by Truscon Laboratories, which is designed to improve the bond between the old and new concrete.

Placing chairs and plates on stringers preparatory to drilling the cinch rod holes.

Concrete for the stringers was poured in the track-occupancy period between 1.30 p.m. and 4 p.m. Mixing was done outside the west end of the tunnel and was hauled to the point of use in special hoppers mounted on two ordinary trailer cars, which were pulled by motor cars. To avoid violating a safety rule concerning the pushing of trailer cars by a motor car, two motor cars were coupled to the trailers —

one at each end. Thus, the concrete « train » could be pulled in each direction.

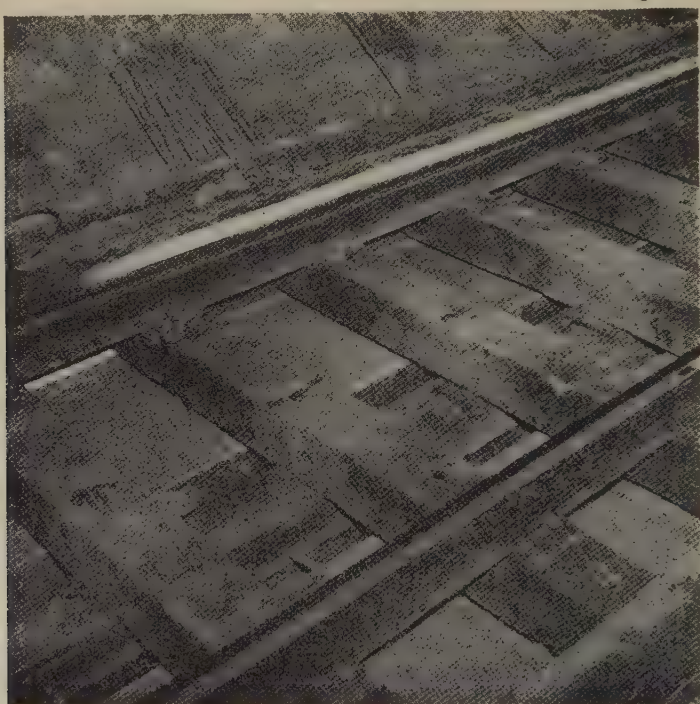
Building the concrete stringers began on November 17, 1948, and was completed on October 7, 1949. The next step was to lower the track on to the continuous steel plates and rail chairs on the stringers. The continuous plates were 20 ft. 3 in. long, 15 in. wide and 5/8 in. thick. The plates were shop drilled with two rows of 1 3/16-in. holes 9 in. apart; these holes had a longitudinal spacing of 1 ft. 6 9/16 in. They were for the bed-plate hold-down bolts. In addition, 1 1/32-in. holes with a 3 in. stagger were drilled and countersunk from the bottom of the plates on the same gage line as the hold-down holes, but located midway between them.

Before the bed plates left the shop, the 1 1/32-in. holes were filled with 1-in. threaded bolts, 5 1/8 in. long. which were inserted from the bottom of the plates and welded in place. The purpose of these bolts is to hold the rail chairs in place as will be shown later.

When installing the plates, short sections of the old track were removed. At first the plates were carefully placed in exact position of the stringers while the positions of the 1 3/16-in. holes were marked on the stringers. The plates were then removed and air tools were used to drill 1 3/4-in. holes into the concrete stringers and into the old concrete, the depth of each hole being 11 1/2 in. As the work progressed it was found that it was simpler

Concrete for the new stringers was mixed at the American end of the tunnel and hauled in special hoppers mounted on trailers to the point of use.

to drill the concrete with the plates in place. Plain cinch anchors, 1 in. in diameter were then placed on 7/8-in. by 14-in. bolts and screwed snug to the bolt head, after which the bolts were placed heads down in the holes.



A section of track in the tunnel showing new stringers before the plates or chairs had been applied.

Fabco pads used.

The next step was to apply the rail chairs. Before this was done, however, cushions consisting of Fabco pads, 6 3/4 in. by 13 in. by 3/8 in., were placed over the short bolts projecting above the bed plate. These pads are a mixture of cotton duck and rubber — actually the « trim » from the manufacture of Fabreeka pads. Their purpose in the tunnel is to serve as a resilient cushion beneath the chairs, thus reducing vibration and cutting down the noise and vibration of train movements. The chairs, which are placed on top of the pads, resemble a tie plate except for a much thicker base and are 6 1/2 in. by 12 in. in plan. The base is 1 1/4 in. thick at the outer edge but only 31/32 in. thick at the inner edge, thus imparting a cant of 1 in 20 to the rail. The chairs were designed for use with 100-lb. rail.

Each chair is drilled with four holes, one near each corner. Two of the holes — in opposite corners of the chair are slotted 1 1/8 in. by 2 in. and are for the chair-holding bolts. The other two are for 7/8-in. by 3 1/2-in. bolts which were inserted from the bottom of the chairs before

installation and which serve to hold a rail clip in place on each side of the rail at every chair. After the rails were lined and gaged, the chairs were bolted securely to the plates.

the corresponding amount of new rail on the chairs. The remaining bolts were installed when time permitted. Hot lead was poured around each cinch bolt shortly after the rail was installed on the chairs,

View of a section of track after work was completed except for placing the guard angles.

after which the plate-holding bolts were made tight. A prefabricated timber runoff, 36 ft. long, was used to overcome the difference in elevation at the end of each day's work.

The installation work was carried out in increments of 160 track-feet. The first step in the work on each increment was to install the plates, complete with chairs, on both stringers with 1/2 of the hold-down bolts applied. The next step was to install

Among the power tools and equipment

used on the project were two Ingersoll-Rand 105 c.f.m. and one 315 c.f.m. air compressors, a 1/2-cu. yd. steam-powered concrete mixer, four jack hammers, 4 concrete busters and a number of impact wrenches.

Toronto, J. W. Salmon, who retired recently as engineer of bridges, exercised general supervision over the project, with E. T. Gove, division engineer at London, Ont., and F. W. Young, bridge and building master, in direct charge. The type of track construction used in the tunnel was developed by C. P. Disney, retired engineer of bridges of the Central Region, who holds Canadian and United States patents on the various elements of the design.

The character and arrangement of the rail-holding clips, and other elements of the special track construction, are seen in this close-up view.

This project was planned and executed under the general direction of E. R. Logie, chief engineer of the Central Region of the C. N.R., Toronto, Ont., and J. W. Demcoe, engineer maintenance of way,

[656 .222 .1]

Recent developments in train speed recording.

Positive action and simplicity of design incorporated in an instrument with a range of 5 to 100 m.p.h.,

by F. R. Axworthy, A.M.I.E.E.

(*The Railway Gazette*, February 9, 1951).

The importance of train speed recorders is emphasised by the fact that British Railways appointed a committee, under the chairmanship of Mr. W. M. Bond of the London Midland Region, to consider the various instruments available and to enquire into the possibility of providing an improved design. Instruments for recording the speed of trains may be installed at the side of the track many miles from stations or other railway buildings. This factor imposes certain limitations in their design. They must be weather-proof, completely self-contained, portable and robust. If they are electrically operated, as is usual, they must be capable of being operated by batteries and their power consumption should be low enough for these to last a reasonable time.

Various methods.

The usual method of obtaining the speed of a vehicle, by measuring the angular velocity of a wheel of known dia., is not applicable when recordings are to be made at a point remote from the vehicle. The average speed of a moving body, being equal to distance divided by time, can be obtained in any one of three ways : By (*a*) measurement of both distance and time; (*b*) by measurement of distance travelled during a fixed time; and by (*c*) measurement of time taken to travel a fixed distance.

Of these, the first is the least attractive since it involves handling two variables. Of the two remaining methods, (*b*) has the advantage that the speed is directly proportional to the measured distance, whilst (*c*), in which the variable is the quantity most readily measured, the speed is inversely proportional to the time.

It follows that an instrument operating on the principle of (*b*) has a scale inherently evenly divided; whereas by use of (*c*), a scale shape of the form shown in fig. 1 *a* is produced. Further consideration of (*b*), however, shows that a practical instrument would be somewhat costly to manufacture. It would almost certainly be electronic in principle and would require specialist operation and maintenance. Thus (*c*) is the method which has been most generally applied to the solution of the problem, and fig. 1 *a* is typical of the scale shapes of early train speed recorders.

Recent developments in the design of instruments of type (*c*) have been mainly centred around methods of improving the scale shape. A recorder developed at the Building Research Station was described by R. S. Jerrett and F. G. Thomas in 1944*. This records the measured speed by drawing lines on a chart and compensates for the inverse law of the scale by using a cam to move the pen. The shape of the cam is calculated to provide a scale that is evenly divided over a range of from 25 to 80 m.p.h. The cam is driven by a synchronous motor, the a.c. supply being provided by a transformer and vibrator unit operated from a 12 V battery.

The Derby research laboratories of the London Midland Region of British Railways have applied the same principle to a speed recorder in which the pen is replaced by a mechanism which prints the speed

(*) *Journal of Scientific Instruments*. July, 1944. Vol. XXI, p. 119.

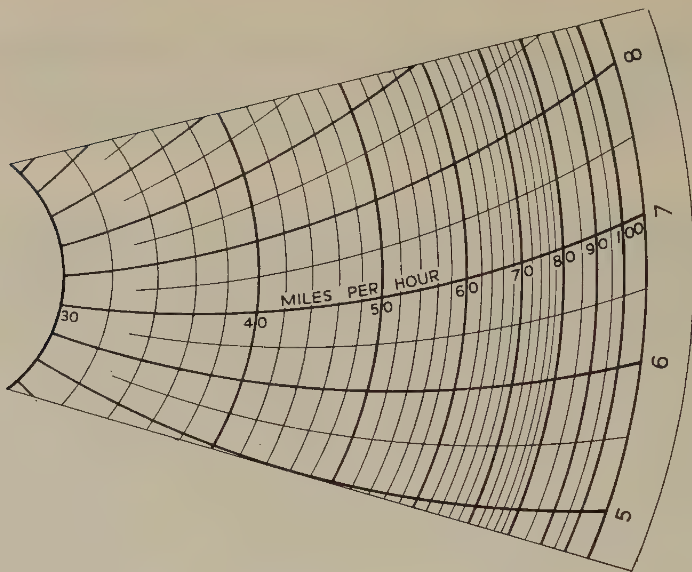


Fig. 1 (*a*). — Scale for operating on principle « *c* ».

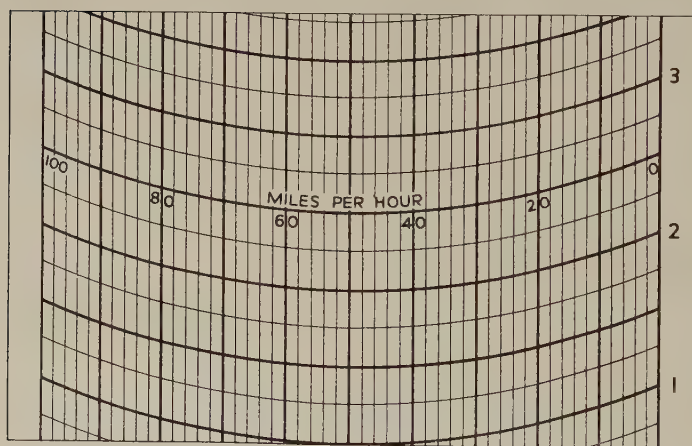


Fig. 1 (*b*). — Scale for operating on principle « *c* » with electrical computer.

of each train, together with the time of measurement, on a narrow paper tape. In this instrument, which has a range of from 5 to 100 m.p.h., the cam is driven by a governed d.c. motor.

A somewhat different approach to the problem of producing a linear scale when measuring the time to travel a known distance, is made in a new recorder designed by Everett Edgcumbe & Co. Ltd., of Hendon. This contains a simple analogue computer, the output of which is a current directly proportional to speed. If the symbols S, D and T refer to speed, distance

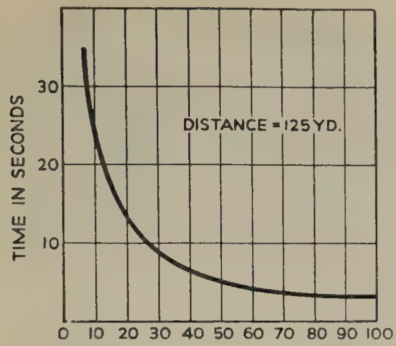


Fig. 2 (*a*).

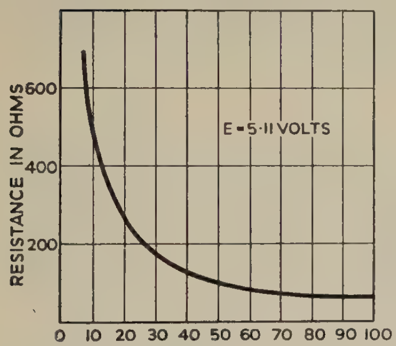


Fig. 2 (*b*).

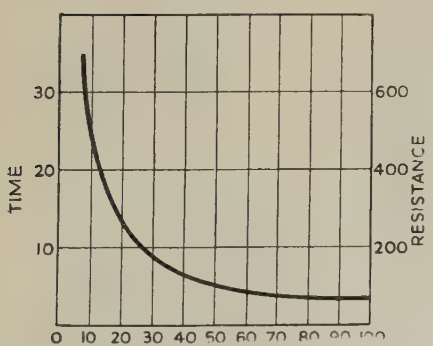


Fig. 2 (*c*).
Relationship between speed, time, and recorder current.

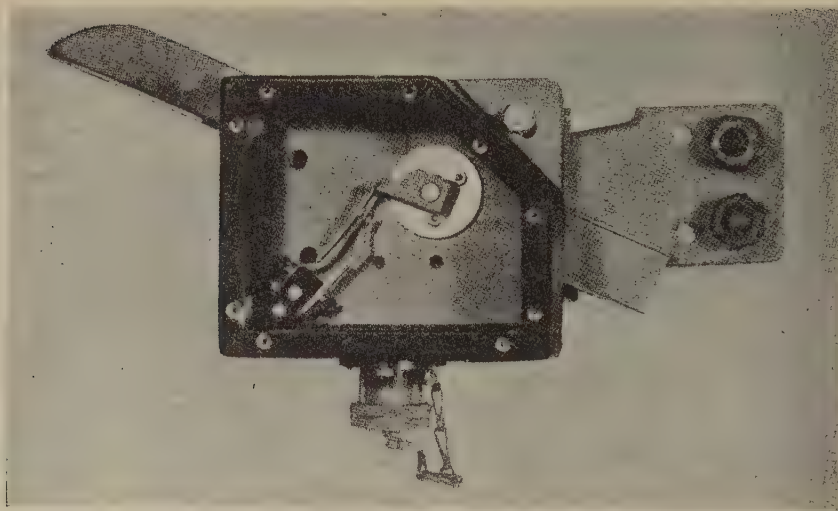
and time respectively, $S = \frac{D}{T}$, which may be compared with the well known formula derived from Ohm's law, $I = \frac{E}{R}$, where I is the electric current flowing in a circuit of resistance R when a voltage E is applied.

If, therefore, the known distance D is represented by a fixed voltage E applied to a circuit whose resistance is made to vary directly as the time, the current which flows will be directly proportional to the speed. The theory of the instrument is illustrated in fig. 2, where (*a*) shows the variation of speed with time over a fixed distance and (*b*) shows the variation of current with resistance when a fixed voltage is applied. With the choice of suitable voltage and resistance values and the latter arranged to vary directly with time, it is seen in (*c*) that the two curves, when drawn to a common scale of current and speed, are coincident.

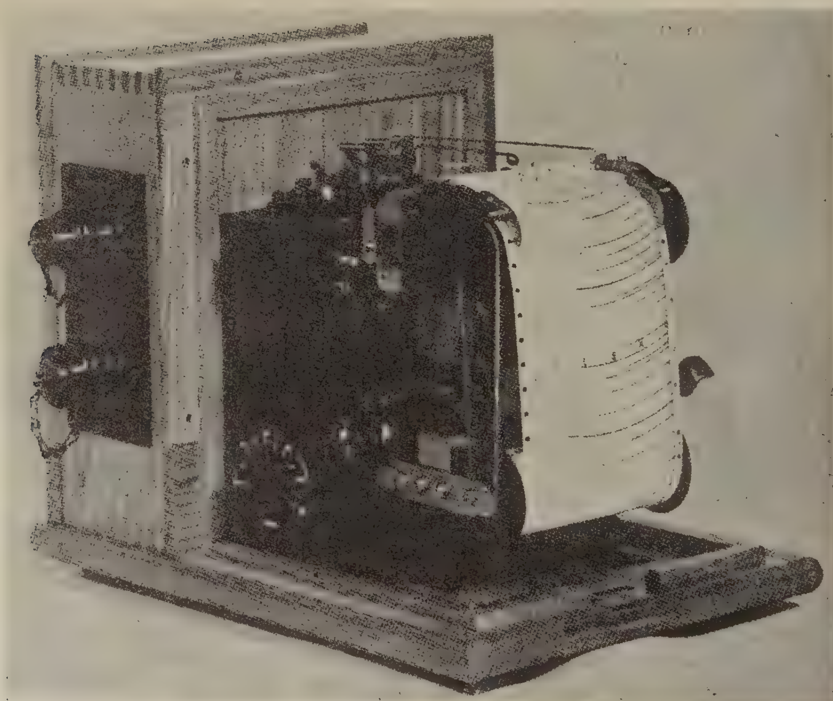
Under such conditions, therefore, there will be equal increments of current for equal increments of speed. In the practical application of this principle, the resistance comprises a rotary potentiometer, the moving contact of which is driven by a synchronous motor energised from a vibrator. The current is measured on a moving coil milliammeter calibrated in m.p.h., the result being an evenly divided scale of speed as shown in fig. 1 (*b*). The computer unit is illustrated on the next page.

It will be realised that the accuracy of the instrument depends on how closely the fixed reference voltage is maintained at its nominal value. It is also affected by the frequency stability of the vibrator, since a change in frequency results in a change in speed of the synchronous motor and hence in an incorrect value of resistance in the computer circuit. However, the interdependence of the reference voltage and computer resistance is used to provide automatic compensation for variations both in battery voltage and vibrator frequency. The method is illustrated in fig. 3 above.

The reference voltage, E, is derived from



Track switch with cover removed.



Track recorder and base length adjuster.

a transformer energised at the same frequency as the synchronous motor. This transformer is designed to be magnetically saturated throughout the normal range of battery voltage over which it is proposed to work. Being saturated, the average secondary voltage, E, is independent of the value of primary voltage, but is directly proportional to the vibrator frequency. It follows, therefore, that the output current is not affected by changes in battery voltage, and that any change in vibrator frequency results in both the voltage and the resistance of the computer circuit changing in the same proportion to maintain the output at its correct value.

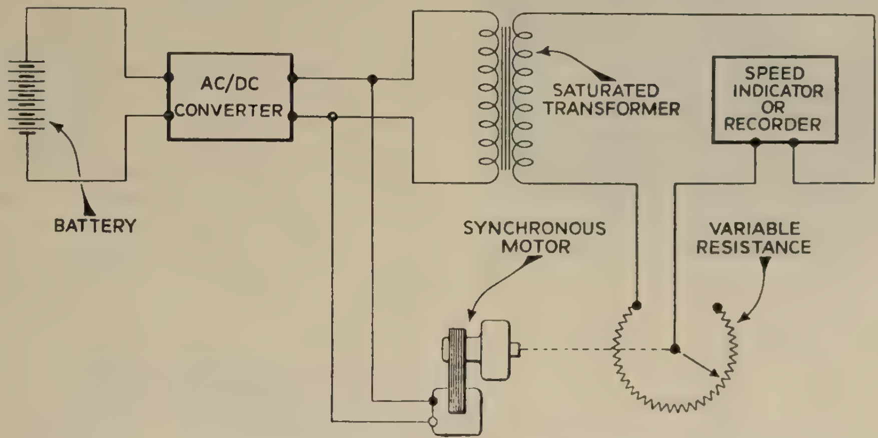


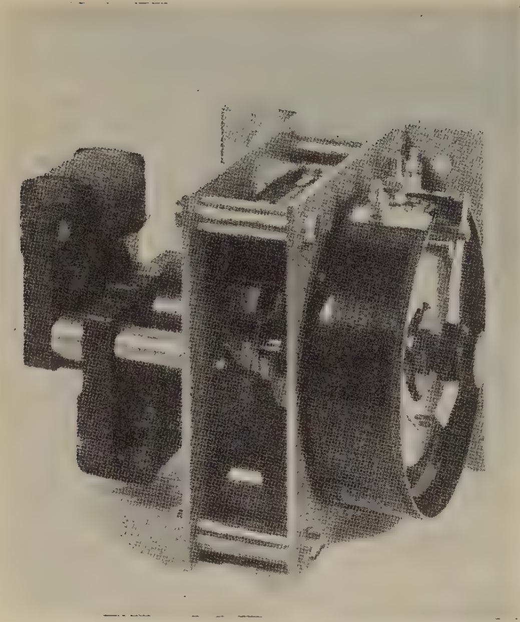
Fig. 3. — Simplified wiring diagram of recorder.

Having an electric current as the output of the computer introduces a high degree of flexibility to the instrument. Speed indication at a point remote from the recorder is easily achieved, another series milliammeter being the only requirement. If the remote indicator is fitted with contacts, all trains exceeding a pre-set speed could be made to give a visual or audible warning. Again, the base length, or distance over which the time is measured, is easily made adjustable with electrical measurement. Reducing the base length reduces the computer resistance at any given speed and consequently increases the current. This can be corrected by altering the sensitivity of the milliammeter. If the sensitivity is made adjustable by means of a continuously variable shunt, the shunt can be calibrated in terms of the base length.

Alternatively, the voltage applied to the computer can be altered to allow for changes in base length. Although either of these methods can be used, the latter has certain practical advantages. The present instrument operates on a timing distance which may be varied between 100 and 200 yds. or between 75 and 150 yds. However, the same computer system could be used for a speed recorder operating on a base length of from 1 to 3 ft., although this would make the instrument somewhat more expensive.

Train operating switches.

The base length is defined by train operated switches on the track. The time of operation of a switch being necessarily short, self-holding high-speed relays are required in the control circuit. These operate further multi-contact relays which perform the various control functions. The circuit is arranged to measure the speed of trains travelling in both directions. It is self-resetting, but will not reset until there



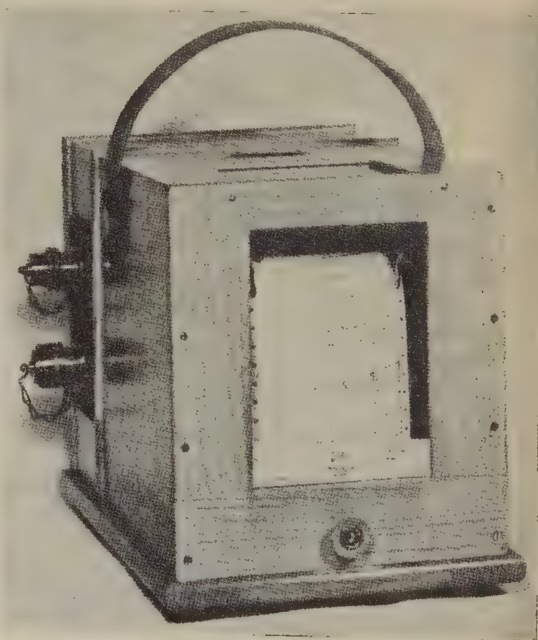
The computer unit of the recorder.

has been an interval of at beast 30 seconds between impulses received from the track switches. This latter feature prevents an incorrect reading being made by a long slowly moving train straddling the switches. To ensure that the motor stops immediately, when the second switch is made, d.c. is substituted for the normal a.c. supply, thus locking the rotor.

If only one switch is closed the recorder will automatically reset after a suitable interval. It will not record the speed of trains moving at less than 5 m.p.h. The equipment operates from a 12 V battery with a power consumption of 18 W during measurement periods. There is no consumption while the instrument is quiescent.

The recording milliammeter is of a standard design of which many thousands have been made for various purposes. It has a permanent magnet moving coil movement developing a very high torque. The pen is of the syphon type fed from a fixed reservoir. The roll chart, 65 ft. in length, is driven by a clockwork motor which includes a lever escapement for accurate time-keeping. Various chart speeds are provided and these are readily changed. With the chart operating at 1 in. per hr. it is possible to differentiate between trains at three minute intervals, but a chart speed of 3 in. per hour may be preferred where the train frequency is high.

Apart from the recorders, the track switches have also presented a problem in design. The British Railways committee on speed recorders put forward several tentative switch designs, and after field trials of the various types, a development of a lever operated switch, originally due to Scottish Region engineers of British Railways, has been produced by Everett Ergcumbe & Co. Ltd. The switch is of light weight, positive in action, and may be adjusted for height after clamping to the rail. The duration of contact closure is amply sufficient for the operation of the recorder.



Everett Edgcumbe train speed recorder.

[625 .143 .3 (44)]

New methods for testing rails by sounding,

by M. PALMÉ,

Ingénieur, Chef de la Subdivision des Rails à la Société Nationale des Chemins de fer français.

(Revue Générale des Chemins de fer, June 1952.)

Preamble.

For some years the use of *supersonics,* in its application to the examination of materials by auscultation, has been notably developed. Since steel rails at times show serious internal defects, which are practically impossible to identify by current methods, we have thought of having recourse to supersonic currents.

We shall give a brief survey of the tests carried out in this connection by the « Laboratoire de la Subdivision des Rails », and of the apparatus which can be used for such purpose with some prospect of securing useful results.

Principle of the method.

The principle applied to the detection of defects by the help of supersonics is as follows : the passage of supersonic currents from one medium to another is accompanied by a strong reflection when the two mediums differ greatly in character. The result is that the slightest discontinuity in the metal will almost perfectly reflect the supersonic beam. In this type of application, piezo-electric quartz crystals (1) are used for the emission and for the reception. The defects are disclosed in a diagram which appears on the screen of a cathode oscillograph; moreover, it must be admitted that a correct reading of the diagram is often difficult.

Tests made by the French National Railways.

a) *Detection of transverse fissures in the flanges of the rails.*

This problem arose as a consequence of delivery of a batch of rails showing transverse fissures in the flange; these cracks being difficult to discern, it thus became necessary to search for them in the rails held in the stock-yard.

1) *Preliminary tests* were made in the laboratory by the method known as « tests by transmission » (fig. 1) : two quartz crystals, one the transmitter and the other

(1) It should be borne in mind that a thin blade of quartz, cut to suit the appropriate crystallographic axis and exposed to an electric field has the property of expanding or contracting according to the direction (sense) of this field.

If then this blade is exposed to an alternating field of a given frequency, it will vibrate at the same frequency as that of the said field. The phenomenon is amplified if the thickness of the blade is designed to oscillate in resonance with the frequency of the exciting field (in practice, this resonance phenomenon is always utilised, as without it the amplitudes would be insufficiently stressed to be noticeable).

Actually, the quartz enables the electric vibration to be transformed into a mechanical oscillation.

On the other hand, in this particular case, one must use oscillations of high frequency, because the higher frequency produces a better directive effort, reduces to a minimum the phenomena of diffraction — which would spoil the precision of localisation — and allows the size of the quartz emitter to be reduced. Moreover, these methods of sounding of metals by means of mechanical vibrations are always well responded to by supersonic methods.

the receiver, thickness 3 mm (1/8″) and dia. 30 mm (1 3/16″), were placed at each end of and facing each other along the section of new rail U 33 (length 1 m [3′ 3 3/8″]) under examination by ausculta-tion. These crystals were moved simultaneously along the surfaces of the two end sections.

The frequency used was 980 kHz. Transmission of the supersonic signal was normal

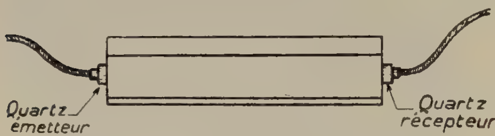


Fig. 1. — Tests by transmission. Position of the quartzes.

Emetteur = transmitter. — Récepteur = receiver.

in the head of the rail, fairly well in the web and in the centre of the flange, but bad at the end of the flange. This was unfortunate as it was the point the most important.

An experiment was made by making a sawcut through *one of the flanges* (fig. 2). This however showed no change so long

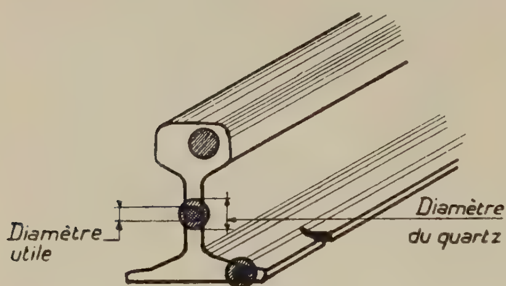


Fig. 2.

Diamètre utile = effective or useful dia. — Diamètre du quartz = quartz dia.

as the sawcut did not extend to a depth greater than 12 mm (15/32″).

A test was likewise made by the method of reflection (the technique being similar to that for Radar), where one uses as a rule only one quartz crystal acting both as emitter and receiver; nothing was observed in the wings of the flange.

The explanation of these failures lies in the *lack of any indication of the course or direction of the emission.* In effect the angle embraced by the beam depends on the ratio between the wave length and the diameter of the quartz : when the latter is placed against the wing of the flange, the supporting surface i.e., the useful diameter is too slight, so that emission loses its directivity.

In connection with this point, a new test was started by changing the thickness of the quartz in order to reduce the wave length thus concentrating the beam [1]. But a quartz of 1 mm thickness was still less favourable, since the absorption at this high frequency was considerable [2] (the absorption of the supersonics increases with the frequency and with a coefficient increasing with a rising percentage of carbon in the metal).

2) New tests were made at the *Levallois Laboratory* with another device, on a sample of U33 rail withdrawn from the track. The quartz crystals used were 2 mm (5/64″) thick by 20 mm (25/32″) dia. The frequency was 1 500 kHz.

In the transmitting tests, the supersonic waves certainly passed through the head of the rail, but not through the web or the flange.

On the other hand, some cracks were disclosed when operating by reflection. But the method adopted was only sensitive to defects which were obviously of importance in themselves.

To summarize, it may be said that the supersonic beam does not transmit readily through a specimen 1 m long when emission takes place in a section of the wing of the flange. There could therefore be no question of sounding rails of 18 m (59′ 5/8″) from the stock pile.

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[1] The thickness of the quartz is calculated as we have pointed out, so that it vibrates in resonance, usually by half a wave length.

[2] About 3 000 kHz.

It should be pointed out that in these experiments, luck was against us, for if it had been a question of disclosing a defect, not in the flange, but in the head of the rail, we should probably have been successful.

b) *Detection of transverse cracks in the head of the rail.*

Actually other tests made at the I. R. S. I. D. (1) were more successful, in this case it was a question of locating defects in *the head* of the rails, in particular of internal transverse fissures (oval stains [marks]).

Tests were carried out by « transmission » and by « *oblique sounding* » (fig. 3) on several specimens cut to a length of 3 m (9′ 10 1/8″). The number of defects disclosed *at that time was considerable,* and this moreover made interpretation of the oscillogram difficult. Nevertheless, the presence of several fissures was verified, by dissecting at the places indicated.

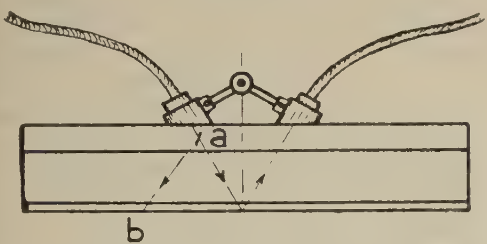

Fig. 3. — Soundings taken by transmission with the help of feelers placed obliquely.

In case of defect such as « a », the supersonic beam is deviated in « b » and is no more received by the quartz.

In all the foregoing examinations we have been troubled by the conditions of contact between the quartz and the steel : actually if the contact is not satisfactory, its surface behaves as if there were a defect, this being due to the principle which is employed in the test.

(1) « Institut de Recherches de la Sidérurgie. »

c) *Use of special feelers.*

In order to avoid erroneous interpretation due to the above conditions, we have been led to experiment with so-called « turgescent » feelers; the emitter and receiver quartzes are enclosed, each in an envelope filled with a liquid and closed (sealed) by a rubber membrane. The liquid gives excellent supersonic transmission and the rubber allows the feeler to mould itself to the roughness (unevennesses) of the part to be sounded, thus avoiding the need to machine the part beforehand, a process which is often necessary with the customary treatment. Unfortunately the tests in this particular respect did not give the results we hoped for.

To sum up, we must say that the tests which have been here described, even those which were successful, have been rather disappointing. The methods employed depend, in fact, more or less closely, on the intensity of the supersonic signal detected or reflected, an intensity which is too much dependent on the perfection of the quartz-steel contact and by the coefficient of absorption of the materials, and this is aggravated by the fact that we are dealing with a hard steel.

This problem of contact is all the more difficult to solve since the head of the rail is curved in section, and because the curvature may vary from one rail to another.

Apparatus for detecting defects in rails.

These considerations and the results obtained have thus, up to the present, militated against sounding rails in the stockyard or on the track by means of supersonic tests. But an apparatus of a new type has just made its appearance, its principle being entirely different, that is to say that it is based on the frequency of the signal and no longer on its intensity.

The apparatus consists in an electronic generator, a vibrating quartz crystal (act-

ing at the same time as emitter and receiver) and includes an earphone, the whole outfit being energized by a small battery of accumulators; these and the generator are fitted into a knapsack. The quartz crystal fixed to the end of a rod is moved about on the head of the rail.

The whole arrangement is calculated so that the intensity of sound heard in the earphone should be a *function of the distance* between the quartz and the first (nearest) reflecting surface met with by the supersonic beam. In principle, the use of this detector is confined to the zone in which is situated the fishing bolts. But in certain cases it has been possible to operate over the whole length of the rail.

The apparatus is operated by moving the quartz along the head of the rail :

— if there is no defect, the flange reflects the waves and a sound of about 1 000 Hz frequency will be heard;

— if there is a horizontal crack in any part of the rail, there will be a premature reflection : the sound heard will then have a lower note.

When the crack is not parallel to the surface of the head, the incident beam and the reflected beam do not coincide; nothing can be heard on the earphones. In this case, the operator cannot judge how far from the surface the crack occurs, but he can discover how long it is by moving the quartz up and down the rail.

It is easy to distinguish between the fishplate holes and cracks and other defects : in fact, the operator always knows where the former are and soon learns to identify the characteristic sound emitted.

The superiority of such an apparatus will immediately be appreciated compared with those mentioned above : detection of flaws no longer depends on the intensity but on the pitch of the sound picked up by the earphones, so that perfection of contact and the coefficient of absorption no longer have the same importance. The signals are easy to interpret and the apparatus can be used by anyone.

Also the results obtained on certain foreign railways appear to be satisfactory, so that we have decided to order apparatus of this kind.

Apart from investigating cracks in the fishplates and testing rails in particular cases, we feel this apparatus should be used conjointly with a detecting apparatus based on the remaining magnetism; it will make it possible to discover the exact nature and position of the defects revealed by the latter.

Summing up.

Supersonics have provided technicians with a very valuable method of testing metals, but unfortunately this does not lend itself very well to an examination of parts of large dimensions and complicated shape — such as rails — made in addition of hard steel, in which the phenomena of absorption are important.

An apparatus based on this new principle seems, *a priori*, exempt from the faults for which its predecessors have been blamed. It is to be hoped that its use, which has proved satisfactory abroad, will give us good results.

[624 .9 & 721]

Notes on certain fundamental principles governing the calculation of frameworks of structures and on the impressions to be gained in the course of reading the two Volumes I & II of « Vorlesungen über Statik der Baukonstruktionen » (Lectures on the Statics of Building Construction), by Prof. Maier-Leibnitz. (*)

Professor Gysen of the Belgian Military College once showed that the fundamental equation for virtual work, as adapted to the study of the stability of structures, allowed the theorem of Maxwell-Betti, as well as those of Castigliano and de Menabrea to be proved as corollaries to that equation.

In France, Bertrand de Fontviolant introduced, as far back as 1906, in his teaching at the Central School of Arts and Manufactures, the general equation for the elasticity of structures based on the same principle, which he called the principle of virtual velocities.

Maurice Lévy, in his treatise on graphic statics, refers to the fact that Mohr, already in 1875, used this principle for determining the tensional stresses in articulated systems with redundant members. It is perhaps as well to recall briefly that a part or system of parts is described as being isostatic or hyperstatic, according to whether pure statics does or does not permit the determination of the mutual reactions of the various parts composing the system. As Professor Gysen points out, the application of the principle of virtual displacements to the hyperstatic cases of stability, is not very general in Belgium, whereas this method is very much favoured in Germany.

At the very beginning of Vol. I, Professor Maier-Leibnitz refers to the latter (p. 2).

In 1864, the well-known scientist J. C. Maxwell published in the *Philosophical Magazine* his thesis « On the Calculation of the Equilibrium and Stiffness of Frames ».

Whereby, he gave to Engineers his famous theorem, known in Belgium as « Maxwell's Theorem », and elsewhere sometimes as the « Theorem of Reciprocal Strain » (*Das Gesetz der Gegenseitigkeit der Formänderungen*) (Maier-Leibnitz, Vol. I, p. 140).

As we know, in the case of two forces (or couples) gradually acting on a body secured by fixed attachments, the effect produced by a unit of the first on the displacement of the second is equal to the effect produced by a unit of the second on the displacement of the first.

This useful theorem enables one to find an elegant solution for the line of influence of a hyperstatic element (moment, shearing stress, axial stress, reaction at an attachment or joint) in a system of attachments which do not require pre-deformation.

As we know, it suffices to take a section with respect to the component, of which it is sought to obtain the line of influence,

(*) *Vorlesungen über Statik der Baukonstruktionen*, by Dr.-Ing. Hermann Maier-Leibnitz, Professor at the Technical High School of Stuttgart. Vol. I and II (8 1/4 × 11 13/16 in.), respectively of 174 and 380 pages, with numerous figures and three plates. Publishers : Franckh'sche Verlagshandlung W. Keller & Co., 57, Pfizerstrasse, Stuttgart. (Price : Vol. I : 24 D.M. — Vol. II : 41.50 D.M.)

in other words, to suppress the resistance which gives rise to this component. Then, introducing in the system a component operating in the opposite sense to that for which we are seeking the line of influence, this latter is given to a certain scale by the elastic curve of the system acted on in this way.

The basic equation is easily ascertained.

Let us consider a group of forces P in equilibrium acting on a part at points m. Under the influence of another group of forces Q, likewise in equilibrium acting at points n and entirely independent of forces P, the forces P will suffer a displacement δ_{mn}.

Let us take as virtual displacements compatible with the attachments, the true elastic strains affecting the part under the influence of the group of forces Q causing in any given section a normal stress N_Q. a shearing stress T_Q and a bending couple M_Q. The general equation for virtual work (we are not assuming a torsion couple) would be written :

$$\Sigma P\delta_{mn} = \int \frac{N_P \, N_Q \, ds}{E\Omega} + \int \frac{T_P \, T_Q \, ds}{KG} + \int \frac{M_P \, M_Q \, ds}{EI}$$

E, Ω, K, G and I having their customary significance. This is the expression which provides the elegant and rapid proofs of Maxwell's and Castigliano's theorems together with their corollaries.

In research on lines of influence in isostatic systems, the elastic strains are replaced by the true displacements of the points in the sectionalized system in accordance with the attachments.

Research on the line of influence of a hyperstatic element shows the need for determining the elastic curve.

One is led to draw the elastic curve of the structure. This part is usually solved in Belgium by a graphic integration, one of the most effective methods, discovered by Massau.

In Germany, Mohr's diagram of the bending moment is commonly applied. It is this latter method that Professor MAIER-LEIBNITZ uses in his works.

In order to terminate this brief survey of the fundamental principles, we may likewise refer to the Hardy Cross method, which was also taught by MAIER-LEIBNITZ. This method is readily applied to hyperstatic structures and tends to replace the method of strain energy, the Gehler method and that of the fixed points. Finally, we may note the recent appearance of a theory leading to a practical solution of these problems.

This method, perfected by the engineers Robert and Musette, comprises in particular :

1) the judicious choice of a reference system, which can equally well be hyperstatic and simpler than that treated, or isostatic;

2) fixing of equation of conditions, which shows that the continuity characterising the true system persists also in the reference. The determination of the values of the coefficients of these equations is readily effected thanks to a formula established by the authors;

3) the resolution of the systems of linear equations by the process introduced by Gauss, a process which can be applied thanks to the symmetry of the elements of the determinant with reference to the main diagonal, this being possible by virtue of Maxwell's theorem referred to above. This process leads to equations which can be solved in stages, their solution is simple, thanks to the choice of hyperstatic variables and to the order of numbering of the same; this order is capable of controlling the structure of the equations dealt with in stages.

This method, now being taught at Brussels University, shows, I think, great promise for the future.

As Professor Baes remarks in his preface to the work published by the above mentioned Belgian engineers : « The lucidity of this method is likely to produce new types of systems of construction, or to make the existing types better-known, thanks to the means which will permit of a complete investigation, enabling all the characteristics to be determined with certainty. »

The present epoch is marked by systems having a high degree of hyperstaticity, whereas hitherto, the engineers went out of their way to avoid them. This is due to the progress made in methods adopted for works in ferro-concrete and in welded metallic structures.

Both in Germany and in Belgium, the theory and design of structures have, in these latter years, had to be modified to suit modern requirements.

Generally speaking, Professor MAIER-LEIBNITZ adopts a syllabus that is practically identical with the Belgian programme.

A very brief résumé of the subject matter dealt with in the first two volumes is given below :

General particulars of the structures.

Principles governing the calculation of isostatic and hyperstatic frames.

Parts subjected to simple bending, supported at both ends.

Determination of shearing stresses and bending moments.

Deflection curves of the structures.

Particular cases of frequent occurrence with examples.

Lines of influence and their coefficients.

Continuous girders having constant and variable moments of inertia, suspension bridges, arched bridges and trusses.

Hardy Cross method.

Simple, multi-span, multi-storey frames and portals.

Numerous examples of research on lines of influence.

What contributes to the originality of this statement, is the number of examples selected with discernment and methodically, leading from the simple on to the complex conditions. The different works clearly presented by the editor « Franckh'sche Verlagshandlung », Stuttgart, should be studied in their entirety, if the practical engineer wishes to enjoy the full advantages of such abundant documentation.

They form a useful guide both for the engineer and the student.

In conclusion, I cannot help thinking of the preface to the French volume : « Problèmes de Résistance des Matériaux » (Problems of the Strength of Materials), published by Albin Michel of Paris, and written by Edouard Callandreau, Professor at the « Ecole Centrale des Arts et Manufactures de Paris ». According to him, the conferences, the preparatory work and tests, the projects do not sufficiently meet the case for the enquiring student : « There is not room for many examples. The collection of problems is intended to teach the students to use the « tool » which has been given to them and to inculcate by practice and perseverance the variations shown by the methods themselves, as also the discipline which is a reflection of the choice or use of the theories or formulae. It is these things which help the student to understand the sense in which this science is applied and at the same time its mode of utilisation. »

I imagine that it was a similar preoccupation, which guided Professor MAIER-LEIBNITZ in the development of his course of lectures.

Everything leads me to believe that he was fortunate in establishing a successful transition between the theory as stated and the problems arising in practice.

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M. LÉVY : « La Statique Graphique ». *Gauthier-Villars*, Paris.

G. MAGNEL : « Pratique du Calcul du Béton armé », 2e partie. *Fecheyr*.

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E. ROBERT et L. MUSETTE : « Le Calcul des Systèmes Hyperstatiques ». *Desoer*.

Below are two figures (reduced size), taken at random, showing the care taken by the publisher of Professor MAIER-LEIBNITZ's work.

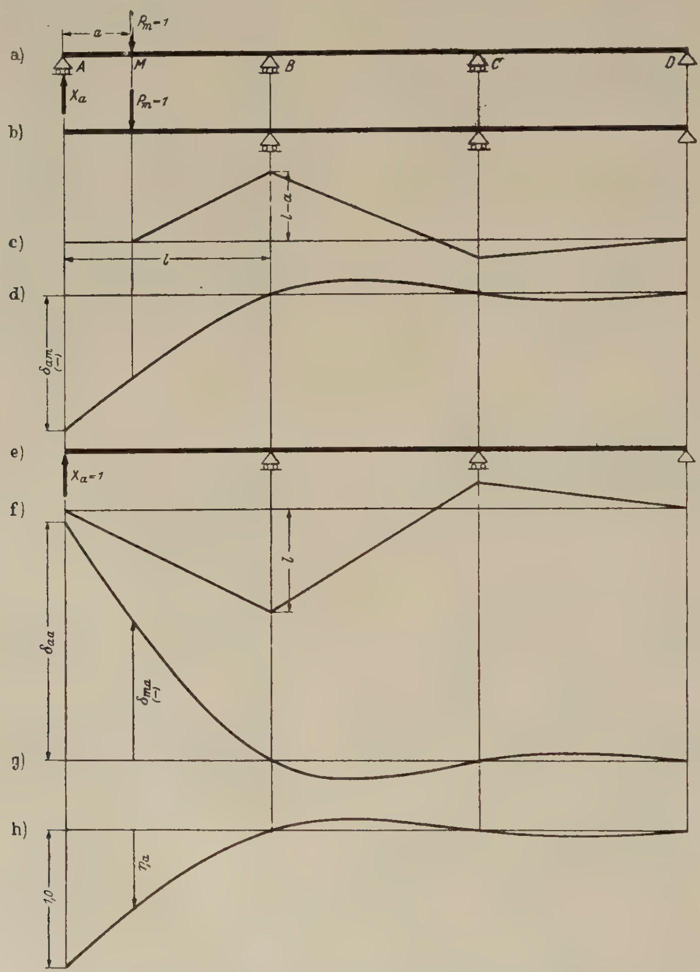

Fig. 10.6 (p. 145, vol. I). — Determination of the line of influence of a reaction of a continuous girder supported at several points.

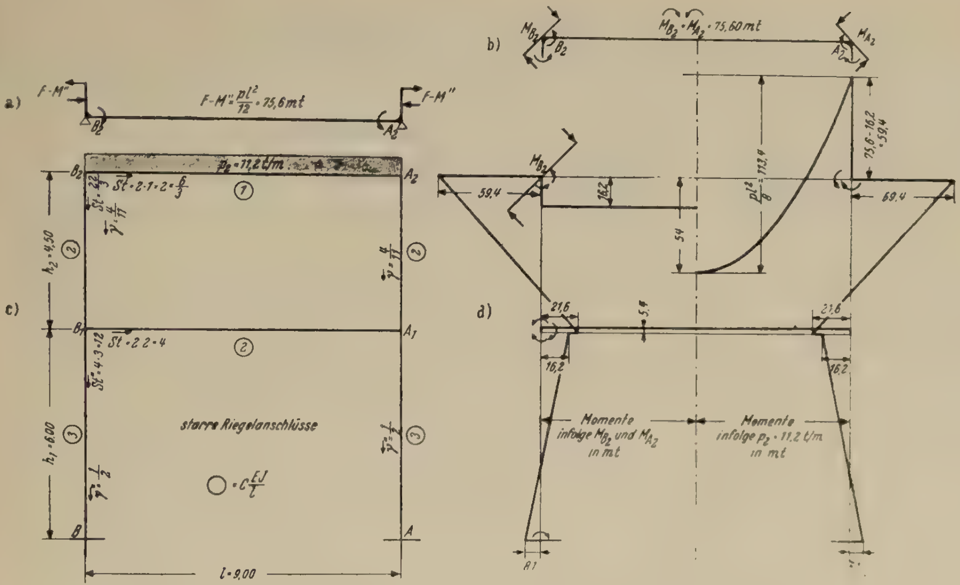

Fig. II.25.16 (p. 341, vol. II). — Double storey portal frame.
Moments due to uniform overload of the upper transom.

Pierre DUBUS,
Ingénieur civil A.I.Br.

[385. (06 .111]

OFFICIAL INFORMATION

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XVIth SESSION — LONDON (1954).

LIST OF QUESTIONS

for discussion

WITH THE NAMES OF THE REPORTERS.

1st SECTION : WAY AND WORKS.

QUESTION 1.

What are the present tendencies relating to the organization of the maintenance of the permanent way : methods of determination of the works to be done and in particular, possibilities of the use of detecting-recording coaches; planning of the works, effects of mechanization; importance of the side-tracks for the movement of the gangs and the mechanical devices.

Economic and financial aspect.

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M. O. Hjelte CLAUSSEN, Ingénieur de la Voie, Chemins de fer de l'Etat danois, 40, Sölvgade, Copenhague K.

QUESTION 2.

Modernisation of station buildings and methods employed in financing modernisation projects.

Standardisation of unit construction applied to railway building.

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Dr. F.F.C. CURTIS, Architect, The Railway Executive, British Railways, 222, Marylebone Road, London, N.W. 1.

2nd SECTION : LOCOMOTIVES AND ROLLING STOCK.

QUESTION 3.

Technical and economic investigation of the basic characteristics of electric traction systems now in use, with a view to decide whether, and to what extent, there are relevant reasons for preferring one system to another.

In particular are there any reasons in regard to :

a) **power supply,**

b) **overhead line and fixed track installations,**

c) **motive power units,**

d) **working and maintenance costs.**

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Mr. S.B. WARDER, Chief Officer (Electrical Engineering), The Railway Executive, British Railways, 222, Marylebone Road, London N.W. 1.

QUESTION 4.

Means and methods to improve the efficiency of steam locomotives :

a) **increase of the steam pressure,**

b) **types of grates,**

c) **superheating of the steam,**

d) **preheating of the feeding water.**

e) **feed water treatment,**

etc.

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Mr. C.T. LONG, Assistant Chief Mechanical Engineer (Motive Power), South African Railways, Johannesburg.

3rd SECTION : WORKING.

QUESTION 5.

Radiophonic communications in railway working.

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Mr. J.H. FRASER, Chief Officer Engineering (Signal & Telecommunications), Railway Executive Headquarters, 222, Marylebone Road, London, N.W. 1.

QUESTION 6.

a) **Remote operating of signal boxes : technical realizations, working orders.**

b) **Electric working and control devices for hinged and « flexible » points and switches. Control of accidental trailing of the switch blade.**

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Mr. SORVIK, Resident Engineer at the Electrical Department of the Executive Offices, Norwegian Railways, Oslo.

4th SECTION : GENERAL.

QUESTION 7.

Modernisation of the methods to be adopted for recruiting the staff in number and qualifications.

Harmonious renewal of the various ranks, indispensable reserve lists, ratio of the permanent and temporary staff.

Part played by the medical service in the recruiting.

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Determination of the principles of geographical and functional organization of a railway system.

Simplification and retrenchment of the administration of railways.

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Mr. Bengt ADAMSON, Superintendent of the Operating Department, Swedish State Railways, Stockholm.

QUESTION 9.

Railway participation in road transport undertakings.

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Mr. TISSOT VAN PATOT (J.P.B.), Chief of the Division Cabinet and General Policy of the Netherlands Railways, Utrecht.

5th SECTION : LIGHT RAILWAYS AND COLONIAL RAILWAYS.

QUESTION 10.

Wear of rails on curves :

a) **running effects of locomotives and motor coaches with motor bogies,**

b) **characteristics of track-laying on curve and details of the rolling stock liable to cause premature wear of the rails,**

c) **results of the investigations made and proposed remedies. Use of rail-lubrication processes.**

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Mr. VENKATARAMAYYA, Chief Structural and Permanent Way Designer, Central Standards Office, Indian Railways, New Delhi (India).

QUESTION 11.

Protection of overhead lines, substations, locomotives and motor-coaches against accidents of electric nature (excess voltage, overloads, short-circuits, lighting).

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NEW BOOKS AND PUBLICATIONS.

[385 .113 (66)]

The administration of the Cameroun Railways. Review by the Management. Year 1951. — One volume (8 1/4 × 10 1/2 in.) of 73 pages with diagrams and numerous tables. Published by the Cameroun Railways.

The Cameroun Railway extends to 504 km (313 miles) divided into two lines, the central line of 307 km (193 miles) and the northern line of 160 km (100 miles), together with a 37 km (23 miles) long branch line. These are all metre (3′ 3 3/8″) gauge lines. Altogether there are 60 stations and halts.

The traffic mainly consists of export goods, timber, bananas, cocoa, palm oil products and coffee. The chief imports are building materials, petrol and parafine, metal products, cars and salt. The most important local traffic is home-produced foodstuffs, timber and livestock.

In 1951, the railway carried a total goods tonnage of 520 000 t amounting to 90 million t/km. The extent of the passenger traffic, can be estimated from the following figures: 1 470 000 passengers carried, 102 million passenger/km.

The interest of the report lies first of all in the general survey with which it begins, then in the opening chapter which describes the general organization. Nearly all the rest of the book consists of numerical tables and graphs. But these documents are accompanied by explanatory notes and commentaries, which make it easy to understand them and stress the evolution of the phenomena analysed.

Like other railway transport undertakings, the management has had to deal with motor competition. This has been met by various measures some of which were inspired by the recommendations of the Railway Congress: the setting up of depots on premises owned by the Company, facilities granted in return for a fidelity contract, liberal policy as regards private sidings and the use of containers. Better turnround of the wagons, more careful maintenance of the stock and the modernization of the rolling stock were other steps that have given encouraging results. There has been a definite increase in both the passenger and the goods traffic.

As regards the traction, savings have been realized at the same time as an improvement made in the service by the introduction of Diesel locomotives and railcars.

The financial situation at the end of the working year shows an excess of receipts over expenditure, with an operating coefficient of 0.94 (0.81 if the renewal annuity is not taken into account). The plus-values of the receipts are due largely to the increase in traffic.

As regards the accounts, mention may be made of the use of a system which enables strict budgetary control and an exact determination of the cost of the different services. Some of these costs are given in the statistics in Chapter VII (Financial position). Various measures have also increased the output of the staff.

E. M.

[385 (09 (44)]

Chemins de fer 1952 *(The Railways 1952)*. — A special supplement published by *Science et Vie*. — One brochure (6 1/4 × 9 in.) of 176 pages copiously illustrated. — 1952, Paris (8e), *Science et Vie*, 5, rue de la Baume. (Price : 200 French francs.)

In these times with their love of rock-bottom formulae, the heading « Organisation and Technique » would be the heading most applicable to the railway. The duties falling upon it are so diverse and so varying that more and more ingenuity is required of its officials. Few other industries make a call upon so many different branches of applied science.

In both these directions, progress was becoming faster and faster before the last upheaval, and since then has been headlong. Is this due to a desire to make up for the time lost during the war years? Others may attribute it to the need for fighting the effects of competition. We prefer to attribute the increasing progress which can be proved from reading this book to the desire to make an undertaking essential to the vital needs of the nation ever more effective and more economical.

In France in particular, as this is the country concerned above all in this case, the changes are many. Without doubt the destruction wrought by the war, painful as it was, left the way clear for a thorough renovation, and the management did not fail, whilst taking the necessary steps to restore the traffic as quickly as possible, to prepare for the final reconstruction upon a new basis. The definition of the general policy and its application were facilitated by the standardization already achieved since the amalgamation of the different companies into the S. N. C. F. The latter as is well known had already pooled their various resources.

In this book, which is a series of studies written by the most qualified authors who have taken an active part in the investigation and execution of the projects described, the most remarkable railway manifestations are dealt with.

The rolling stock has supplied the most copious matter, owing no doubt to its great diversity and also because as far as the public is concerned, it is the most visible, interesting and characteristic part of the undertaking. And in the case of the vehicles, it is the engines which receive the most attention. The use of all available sources of power, the ascertaining of the greatest efficiency and its most suitable application for railway traction has resulted in a great variety of engines which have been more and more improved and an ever increasing scale of power.

As regards speed, one of the great necessities of our days, the design of the rolling stock also plays its part. Neither safety, comfort nor economy have been overlooked.

With increasing loads and speeds, the track has to stand up to a lot. Two studies show how it is doing so owing to the way it is made and by reason of careful and well equipped maintenance.

Though less obvious, the internal life of the railway, is equally arresting. To profit by all the resources made available by technical progress for the greater convenience of the public, means carrying out researches as difficult as they are varied. These include the preparation of the timetables, the composition of the trains, the transport of heavy goods and parcels traffic, the equipping and working of the goods stations and marshalling yards.

Finally, the latest technical marvel, electricity, has given safety, which as everyone knows beats all records on the railway, new weapons in operating and control equipment, the performance of which is as remarkable as the simplicity achieved in all their operations. Based on telecommunication gear, the railway control organisation

makes it possible for the management at every degree to see the state of the traffic at any instant and thereby supervise and direct it.

Doubtless there are gaps in this rapid summary of the contents of a book which in spite of its amplitude has not exhausted the subject. For a synthesis written by an authoritative pen, the editorial note by M. Louis ARMAND, the distinguished general manager of the S. N. C. F., should be read.

E. M.

[**621** .13 (02]

CHAPELON (André), Ingénieur des Arts et Manufactures, Chef de la Division des Etudes de Locomotives à Vapeur de la S. N. C. F., Maître de Conférences à l'Ecole Centrale. — **La locomotive à vapeur** *(The steam locomotive)*. — Second edition. Volume one. — One volume in -4° (8 1/4 × 10 1/2 in.) of XVI-648 pages, with 408 figures and VIII plates. — 1952, Paris, Librairie J. B. Baillière et fils, Publishers, 19, rue Hautefeuille. (Price : 7 000 French francs.)

In view of the way various methods of traction have developed, such as electric traction, railcars, Diesel-electric locomotives, without mentioning other inventions under trial, there is a temptation to underestimate the place still held by the steam locomotive. Not only is it still the most widely used method of traction, and not merely in secondary services, but in the last ten years it has made really astonishing progress, which is a threat to some of the positions held by its competitors.

Increased power, improved efficiency, greater velocity, these are the improvements which first of all attract ones attention. But there are a great many other innovations which in particular have increased the strength and reduced the vulnerability of a well proved engine. These measures, in conjunction with new service methods and reorganization of the sheds, have made it possible to achieve a « coefficient of presence » and daily mileages which make it necessary to reconsider certain comparisons made rather hastily.

Another remarkable fact, these excellent results have been obtained without having recourse to any new principles and without going outside the framework of the classic locomotive but as EDOUARD SAUVAGE says in his preface, solely by simple means. These are a result of the methodical and persevering study of the working of both the boiler and the engine, and the two in conjunction.

The author has been the champion in France of a sort of revival of the steam locomotive. He has explained and put into practice the steps whose application proved so profitable. In this book, he gives a report of the investigations and researches undertaken relative to the creation of numerous prototypes on which the new ideas were tried out simultaneously or separately. It is therefore not a didactic treatise but a very instructive book. The tests at the bench and on the line, as well as the results obtained in service are methodically compared and discussed. This led to the locomotive stock being enriched by more powerful locomotives, which were faster and more efficient, to the profit of the railway and above all the public. In reality, we are assisting at the evolution of the steam locomotive over a quarter of a century.

The first edition appeared in 1938. The fact that it was soon out of print is a witness to the success it enjoyed with both builders and operators of all countries.

The second edition will consist of two volumes. It was difficult to condense into a single volume such abundant material. Many new types have been designed and important data recently obtained throw open the door to further possibilities.

This first volume is devoted above all to considerations of a general order. Two orders of ideas can be distinguished in the transformations reported. First of all, there is the progress made in the production and use of the heat, with the consequent increase in the loads and speeds and the reduction of consumption. Then there are the constructive arrangements which have completely altered the appearance of the locomotive, and the methods which have enabled the mileage run to be considerably increased.

We will not try to sum up this first volume here. It will be sufficient to call attention to the chief points on which the research work made to improve the output and the power has been concentrated. These are : the gas and steam circuits, the draught, the distribution, superheating, and compounding.

The author also gives the leading dimensions and characteristic features of a large number of types of locomotives, with the most remarkable results obtained with some of them.

As for still more significant performances, especially as regards loads and speeds, these are the subject of a special chapter. Here will be found mention of the record of 202 km (125 miles)/h achieved on the English L. N. E. R. with the « Mallard » *Pacific* locomotive.

Volume II will be devoted to more theoretical aspects. It will deal with the theoretical and experimental design of the steam locomotive considered as a producer and user of heat and as a tractive vehicle on rails. Like the volume under consideration, it can be predicted that it will be warmly welcomed by all those interested in the steam locomotive and railway traction in general.

E. M.

[**656** (492)]

TISSOT van PATOT (J. P. B.), Chef de Division des Chemins de fer Néerlandais. — **Het Concentratie-Verschijnsel in het binnenlandse Vervoerwegen.** Een inleidende studie. *(The phenomenon of concentration in inland transport. An introductive study.)* — One volume (6 1/4 × 9 1/2 in.) of 182 pages and 1 map.

In this work the author presents a disertation written to obtain his degree of doctor of economic science at the Dutch University of Rotterdam.

The subject in question is of particular interest because in Holland, perhaps more than in other countries, rail transport undertakings at a certain epoch were very scattered. Amalgamations took place however which led to a great simplification in the services. A study of the circumstances which preceded and coincided with this concentration and of its consequences appears all the more useful as there is very little literature dealing with developments of this phenomenon in the author's country.

The latter does not pretend to have made a complete examination of this question. His object was rather to pave the way for a more thorough investigation.

The book is divided into five chapters.

In the first chapter the author examines the unfolding of the phenomenon of concentration considered from the general point of view.

In the second chapter, he characterises the transport undertaking in relation to the economic field in which it exercises its activity. Here concentration takes on a special aspect under the influence of various factors.

In the third chapter, the railways give rise to multiple manifestations of such concentration. The influence of the

intervention of the public authorities on the functioning of the undertakings is lightly sketched in and a brief history given of the amalgamations which took place on the railways in Holland. The factors which determined this evolution are sometimes economic, and sometimes political.

The fourth chapter gives a similar study in connection with the bus services between towns.

The work ends with a résumé followed by final considerations. The latter stress the consequences of the concentration achieved and show to what extent the aims in view have been attained.

E. M.

[**385** (09 (436)]

KARNER (Dr. F.), Manager of the Press Department of the Austrian Federal Railways. — **Was weisst Du von den Bundesbahnen?** *(What do you know about the Federal Railways?)* One brochure (6 × 7 7/8 in.) of 18 pages copiously illustrated. — Vienna, VI. Zeitschriftenverlag Ployer & Co., Ägidigasse, 5.

The general public usually does not know a great deal about the circumstances in which a railway works. Even if it appreciates the quality of the services offered, the price of which it nevertheless considers too high, it often ignores the work accomplished and the difficulties surmounted in the interests of the community. The multiplicity of the tasks which fall upon a railway undertaking and the services offered, so diverse and variable, are too little known. Unfounded criticisms are often due to this lack of information and may have unfortunate consequences.

The very worthy object of this publication is to counteract a mentality which risks undermining the good name of the railway and even its prosperity. Unlike technical periodicals whose readers will be railway professionals so that they will not have the desired effect, this is addressed to the general public. It has obviously been designed to attract the greatest possible attention and be read by those who know the least about the subject.

The author has attracted and held the attention of his readers by pictures. The pages consequently are mainly filled with photographs. The text is reduced to the minimum necessary to give a suitable commentary or the necessary explanation.

In Austria, the destruction caused by the war necessitated very considerable reconstructional work. At the same time the permanent way and buildings as well as the rolling stock were brought up to date. If to this is added the progress made in electrification and the mountainous character of many of the lines, it will be understood that the author had a wide choice in collecting the illustrations for this little book.

One of the last pages meets a reproach often levelled at the railways, namely their unfavourable financial situation. Graphs show the chief economic index figures for May 1952 together with the increases in the rates, compared with the figures for March 1938.

E. M.

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